IN TWO SECTIONS—SECTION TWO

TRANSACTIONS

of The American Society of Mechanical Engineers

SOCIETY RECORDS—Part II

(Part I of Society Records for the year 1935, containing Council and Committee Personnel and other general information, was issued as Section Two of the Transactions for February, 1935)

Calvin Winsor Rice (*Portrait*) RI-36

Life of Calvin Winsor Rice RI-37

Dr. Calvin W. Rice's Contribution to International Friendliness . . . . *John H. Finley* RI-40

Memorial Notices RI-42

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Memorial Notices

THIS group of memorial notices contains records of members of the Society whose deaths occurred prior to 1935, but concerning whom material was not ready for publication when the 1934 Transactions was issued. In fact, it is still necessary to delay the publication of obituaries of a number of members who died in 1933 and 1934 because sufficient information has not been secured.

There are a number of sources from which biographical material concerning engineers may be secured, but the Transactions of the A.S.M.E. is naturally the only place in which obituaries of all deceased members of the Society appear. Special care is therefore given to the preparation of these memorials and no obituary is published until as complete a record as possible has been secured.

The first source of information is the application file of the Society. In the case of those who have become members during recent years, the applications usually yield fairly complete records. The applications of those who became members in the early days of the Society are not as detailed, however, and these are the cases which often present considerable difficulty. The member may have been retired for some years prior to his death, so that business associates cannot be located, and in some cases members of his family cannot be found. Only by following up every slightest lead can even the main facts be assembled, and a long period may therefore elapse between the time of a member's death and the completion of his memorial.

The Society appreciates the assistance that has been given by relatives, business associates, and friends in the preparation of these memorials. It also acknowledges its debt to such sources as Who's Who in Engineering, Who's Who in America, and similar publications; the National Cyclopedia of American Biography and New International Year Book; the technical and daily press; and to engineering and other societies which have supplied information from their records.

Relatives, business associates, and Local Section and Student Branch officers are urged to notify the Society promptly of the deaths of members. Newspaper clippings or obituaries in any other form should be sent whenever available and the names and addresses of those who can supply further information should be furnished. A special form for supplying complete details will be forwarded by the Office of the Society upon request.

Harris and Ewing

CALVIN WINSOR RICE, 1868–1934

Secretary of The American Society of Mechanical Engineers
1906–1934

Life of Calvin Winsor Rice

STRICKEN at his desk in the midst of his daily work, and in the building which his vision, enthusiasm, and energy had greatly helped to bring into being as a headquarters for the professional engineering societies of the United States, Calvin Winsor Rice, honorary member and for twenty-eight years secretary of The American Society of Mechanical Engineers died as a result of cerebral hemorrhage on October 2, 1934, a few hours after his removal to a nearby hospital.

Although he had won distinction as an electrical engineer in the practice of that profession prior to 1906, he will be most widely remembered for his services to The American Society of Mechanical Engineers as its secretary and for his vision and leadership in that office, which not only developed the Society greatly in numbers and in virility, but the entire engineering profession as well. For the spirit of cooperation with which he worked brought together the numerous individually organized groups of engineers in joint activities for common purposes. Of this spirit, the Engineering Societies Building in New York is a substantial and glorious monument, the physical embodiment of that vision and idealism that were Dr. Rice's most abundant gifts.

His Qualities in Professional-Society Organization

At the time Dr. Rice was made an honorary member of The American Society of Mechanical Engineers, in December, 1931, Dr. Karl T. Compton, president, Massachusetts Institute of Technology, delivered an address on his work in professional-society organization and paid a splendid tribute to this phase of his career. In the address, which was published in the January, 1932, issue of Mechanical Engineering, Dr. Compton listed six essential qualifications for leadership in engineering-society organization which Dr. Rice possessed to a marked degree: (1) The ability and desire to cooperate with others; (2) the ability and judgment to recognize a good project when it was suggested; (3) initiative and drive to carry a project through to completion; (4) daring to undertake an audacious project, once convinced of its merit; (5) originality of thought; and (6) ability to organize, to delegate authority, and to spur others on to take active part in the affairs of the Society. Dr. Compton listed in addition four principles that were basic to Dr. Rice's philosophy of society organization and operation: (1) Unselfish cooperation with related societies; (2) planning actively for the future, so that development might not be haphazard, and so that opportunities for development in the desired direction might be quickly and firmly grasped when they presented themselves; (3) giving the most effective service to the members of the profession; and (4) leading the profession in rendering the most valuable possible service to society. The great number of engineers who knew Dr. Rice were conscious of these elements of his character and these principles of his philosophy.

His International Reputation

One characteristic feature of his career, resulting from these personal qualities, brought distinction to him, to the society he served, to his country, and to the profession of engineering. This was his relationship with engineers in this and other countries. In Great Britain, in Europe, in Mexico and South America, and in the Orient, Dr. Rice was well known in engineering circles. He traveled frequently in the discharge of his duties, and his office in New York was a focal point upon which converged the paths of engineers, eminent and obscure, who came to the United States. He was never too busy to give letters of introduction to engineers from abroad, or to engineers of this country planning to travel in Europe. Members of the Society frequently found that letters from Dr. Rice were more effective in gaining for them admission to some foreign plant or factory than similar letters from business men and engineers in the country they were visiting. The bread of hospitality and service he continually cast upon the waters wherever he happened to be returned in abundant measure to him by way of courtesies shown all over the world to those for whom he bespoke consideration. In students coming to this country Dr. Rice had great interest, going out of his way to have arranged for them itineraries of plant visits and advising them on educational and professional programs. These active international relationships brought Dr. Rice into high esteem, both here and abroad, and year by year added to the list of those beholden to him for help, advice, and innumerable lesser courtesies.

Significant Events in His Career

Dr. Rice was born at Winchester, Massachusetts, on November 4, 1868, the son of Edward Hyde and Lucy J. (Staples) Rice. After attending public schools in Boston, New Haven, and Winchester, he spent four years as a student in the Massachusetts Institute of Technology, from which he was graduated in 1890 with the degree of bachelor of science in electrical engineering. He then held, successively, positions as assistant engineer in the power and mining department of the Thomson-Houston Company in Lynn; as engineer in the General Electric Company in Schenectady; district engineer for that company in Cincinnati; engineer with the Silver Lake Mines in Colorado; consulting engineer for the Anaconda Copper Mining Company in Anaconda, Montana; electrical engineer of the Kings County Electric Light and Power Company, and later with the New York Edison Company and the Consolidated Subway Company; vice-president of the Nernst Lamp Company; consulting engineer with the General Electric Company in New York. It was from this rich and varied experience in electrical, hydraulic, and steam engineering, combined with managerial and administrative work, that Dr. Rice was called to the secretaryship of the A.S.M.E. in 1906.

The Engineering Societies Building in New York

Dr. Rice became a member of the A.I.E.E. in 1897, and was active in its affairs. In 1900 he joined the A.S.M.E. The story of his part in the bringing together of the engineering societies in a common headquarters building in New York and of the manner in which the funds were secured was told by Dr. Compton in his tribute to Dr. Rice previously referred to. Dr. Compton said:

> In 1902, as chairman of the Building Committee of the A.I.E.E., he called a dinner meeting of the committee, together with the president of the A.I.E.E., Prof. Charles F. Scott, and several others, to discuss plans for a modest building for the Institute, primarily to house the Latimer Clark Library, which had been presented to the Institute by Dr. S. S. Wheeler on the condition that a fireproof building be secured to house it. The Committee had, at that time, definite prospects of only about $250,000. When President Scott suggested that consideration be given to the possibility of a building for housing the four National Engineering Societies, with a common library and a common auditorium and individual rooms for the headquarters of each society, doubts were expressed as to whether the four societies could be brought into such a cooperative project.
>
> Strenuous efforts, which were at the last minute successful, were made to get Mr. Andrew Carnegie as a guest at the next annual dinner of the Institute. At this dinner President Scott outlined his ambitious plan and pointed out its fine features—including the library.
>
> The next day Mr. Carnegie asked Dr. Rice to come to his residence at five o'clock, and Dr. Rice, with characteristic thoughtfulness for others, as well as admiration for the lofty character of his president,

brought with him Professor Scott. There Mr. Carnegie asked them further about the work of the Institute, about the finances of the engineering societies, about the relation of the proposed building to the Engineers' Club (of which he was a member). Dr. Rice cleverly inferred that an obstacle in Mr. Carnegie's mind was the securing of the land, for the latter was not in the habit of buying the land on which the libraries which he donated were built. Dr. Rice then optimistically remarked that the Engineering Societies would be able to provide the land, whereupon Mr. Carnegie gave a cheerful smile and said, "If you can provide the land, I will put up the building." Dr. Rice was made chairman of the building fund.

Then the money had to be raised to buy the land; complications and difficulties in perfecting the organization and developing the plans had to be overcome. In the words of Professor Scott, "Mr. Rice's devotion to the idea of a building for the Institute and his skill in directing the early conference with Mr. Carnegie and his enthusiastic and faithful assistance in subsequent service to the Institute in carrying out the project were fundamental factors in the creation of the Engineering Societies Building and the separate building for the Engineers' Club."

The Kelvin Memorial

Among other incidents that illustrate Dr. Rice's faculty of initiating projects that bore fruitful results may be mentioned the Kelvin Memorial Window, in Westminster Abbey, told by Dr. Compton in the A.S.M.E. address, and the establishment of the Officers' Reserve of the United States Army, a brief account of which was contained in a letter by the late Gen. William Barclay Parsons published in *The Military Engineer* for March-April, 1931. Dr. Compton's account of the Kelvin Memorial Window is as follows:

In 1910 Dr. Rice had a most unique experience, an account of which has never before been published. That year the A.S.M.E. made a return visit to the Institution of Mechanical Engineers at their Birmingham meeting. Remembering that in a modest way the A.S.M.E. had contributed to the memorial window in Westminster Abbey to Sir Benjamin Baker, honorary member, A.S.M.E., Dr. Rice wrote to the Dean asking if it would be permissible for the members of the A.S.M.E. when passing through London on a certain Sunday, to visit the Abbey and view the window. Not only was permission granted but a special service was arranged, with a sermon on engineering by the Bishop of Lewes, and on this occasion the "Hallelujah Chorus" was rendered by the full surpliced choir. The event was further made memorable by having the A.S.M.E. audience arranged in a semi-circle about the memorial window, the movable pulpit having been placed beside it.

Dr. Rice noticed that every window in the entire Abbey, save one, was a memorial window. The unappropriated window was apparently an original plain-glass window and was very dull by comparison. The next day Dr. Rice called on the Dean to express gratitude, and in conversation commented on the unoccupied window. The Dean immediately responded that the Abbey would appreciate a gift of a memorial window. Dr. Rice thereupon sensed the situation and offered a window, knowing it would be an easy matter to collect from the entire English-speaking world an amount sufficient to install a window to an engineer.

Dr. Rice proposed a window to his friend Lord Kelvin as one mutually desired by the Abbey and by engineers. Consistently he arranged that this memorial be provided through the cooperation of the engineering bodies of Great Britain and the United States. Having obtained instant approval of influential persons in England, he used the same method in the United States, and, when the undertaking was assured, placed the whole proposition in the hands of the Institution of Civil Engineers, the oldest and most important engineering organization in the world, for announcement of the popular subscription.

The result was so successful that not only was the window provided but the Kelvin Medal was founded. This is probably the only joint undertaking of this nature by the English-speaking world.

The Officers' Reserve

In a letter to General Parsons, Dr. Rice said that he had commenced agitation for an Officers' Reserve in 1902, when he went to Washington to take the matter up with Gen. Nelson A. Miles, then Chief of Staff. He was unsuccessful, but he kept repeating his suggestion up to 1914. From General Parson's letter in *The Military Engineer* the following passages are quoted:

By the end of the year 1914, there were some engineers who perceived that the world was in for a long period of war and that no matter what position of neutrality the United States might for a while assume, it would probably inevitably be drawn into the conflict. Mr. Calvin W. Rice, secretary of The American Society of Mechanical Engineers, had this view and became firmly impressed with the belief that the engineers should be organized for action.

After conferring with Major-General Leonard Wood, then commanding the Department of the East, Mr. Rice organized a luncheon in February, 1915, at which were present, besides Mr. Rice, Dr. Henry S. Drinker, President of Lehigh University, Mr. Elmer E. Corthell, Mr. Ralph Mershon, Mr. Bradley Stoughton, and a few others, with General Wood as the guest of honor. On being asked for advice, General Wood pointed out that for many years there had been established, as part of the Army, a Medical Officers' Reserve Corps, who were duly commissioned in the Army and who were subject to call to duty in case of war and in times of peace with their consent. This Reserve had been found to be most beneficial, as officers were drawn from it at times of emergency and then could return to private life when the stress was passed. He showed that engineers, like doctors, were always mobilized for the practice of their profession and that an Engineer Officers' Reserve, parallel to the Medical Officers' Reserve, might be established.

This suggestion of General Wood's was promptly taken up, and committees were appointed on behalf of the engineering societies. . . . These committees were immediately organized and began to work, but it was found more expedient to form a central executive and operating committee of the chairmen of the separate committees. . . . The Chairman of the Joint Committee [Wm. Barclay Parsons] and the Committee proceeded to Washington to lay the proposed plan before the War Department. The Committee went with some misgivings as to how such a radical suggestion, emanating from civilians, would be received. Their misgivings, however, were quickly dispelled. The Secretary of War, Mr. Lindley M. Garrison, received them cordially and sympathetically and presented them to Major-General Hugh L. Scott, then Chief of Staff. General Scott, after hearing what the chairman had to say, introduced him to Major-General Tasker H. Bliss, Assistant Chief of Staff. General Bliss listened attentively and with much interest, and at the conclusion of a long interview sent for Major W. D. Connor, now Major-General Connor, an officer of the Corps of Engineers attached to the General Staff. General Bliss instructed Major Connor to confer with Mr. Parsons and to prepare a joint report for submission. When this was done, General Bliss studied it carefully and said, "You have proved your case for engineers, but why limit it to them? It has always been my wish that there should be established a general reserve as part of the Army of the United States." He then returned the report, asking them to make a plan for such a general reserve. . . .

The outcome was that the joint committee reported to the five societies under date of June 23, 1916, that "a bill, which recently passed the Congress, has been signed by the President and will become effective July 1, 1916. This bill, known as the Army Reorganization Act of 1916, contains provisions for the organization of an Officers' Reserve, including the engineers."

Thus a movement initiated by Mr. Rice, put into concrete form by General Wood, and carried into execution by the Committee of the Engineering Societies, was authorized by law.

Dr. Rice, in his modest manner, was very proud of the part he played in the establishment of the Officers' Reserve. He was commissioned a major in the Officers' Reserve Corps in 1922 and a Lieutenant-Colonel in 1929.

His Educational Affiliations

Inspired by the work and character of Dr. Oskar von Miller, distinguished director of the Deutsches Museum, for whom he had a warm affection and high regard, Dr. Rice became interested in the educational possibilities of exhibits of the industrial arts. This brought him into active working contact with a group of men of enthusiasm and vision who established what is now the New York Museum of Science and Industry. As honorary secretary and a member of the board of this institution he gave freely of his energy and advice up to the time of his death.

To his Alma Mater he gave his services as member of the Corporation of the Massachusetts Institute of Technology and chairman of the visiting committee of the department of mechanical engineering of that institution. His broad acquaintanceship

among engineers brought him into close contact with other institutions, for his advice was frequently sought when a professorship, a deanship, or a presidency was to be filled.

Honors and Professional Contacts

In addition to membership in the A.S.M.E. and the A.I.E.E., of which he was a former vice-president, Dr. Rice was a member of many other engineering and professional societies. He was an honorary member of the association of members in Argentina of the National Engineering Societies; of the Koninklijk Instituut van Ingenieurs, of Holland; of the Club de Engenharia, of Rio de Janeiro; of the American Society of Safety Engineers; of the Masaryk Academy, Czechoslovakia; and of the Deutsches Museum, Munich, Germany. He was corresponding member of the Instituto de Engenieros de Chile and of the Technisches Museum, of Vienna. In 1915 he served as a member of the Jury of Award of the Panama-Pacific Exposition. In 1922 he received a gold medal at the Centennial Exposition of Brazil. On him were bestowed the Order of the White Lion of Czechoslovakia, and the Golden Ring of Honor of Bavaria. He was a fellow of the American Association for the Advancement of Science, and a member of the Institution of Electrical Engineers, of London, and of the New York Electrical Society. His well-known interest in research won for him membership in the division of engineering and industrial research of the National Research Council, and the position of National Counselor of the Purdue Research Foundation.

As a delegate of The American Society of Mechanical Engineers to the Seventy-Fifth Anniversary of the Verein deutscher Ingenieure, held in Cologne, in 1931, he was the recipient of a medal of honor "in appreciation of his services to technical-scientific achievement, particularly in promoting the mutual international interests of the engineers of the entire world." Among other honors in Germany he received the honorary degree of Doctor of Engineering (Dr.-Ing. E. h.), from the Technische Hochschule, of Darmstadt, Germany, in 1926.

Dr. Rice married Ellen M. Weibezahn, of Winchester, Mass., August 6, 1904, who, with his children, Edward Winslow and Marjorie Charlotte, survives him.

Tribute of the A.S.M.E. Council

A tribute to Dr. Rice was presented at the Business Meeting of the Society in December, 1934, in behalf of the committee of three past-presidents, Dexter S. Kimball, chairman, D. S. Jacobus, and Alex Dow, appointed for this purpose at the October 24 meeting of the Executive Committee of the Council of the Society. The tribute follows:

In the passing of Calvin Winsor Rice this Society has suffered an irreparable loss and many of its members have lost a kind friend and a wise advisor. To some of us who have been close to him for many years it does not seem possible that we shall never again see his cheerful smile, never again feel the hearty clasp of his hand, and never again see his bold signature at the end of a letter.

The accomplishments of his long and busy life are well known and will be recorded elsewhere, but it is very fitting that we should here be reminded of the great work that he performed in building up this Society. In season and out of season he labored wholeheartedly, and much of the success and reputation of The American Society of Mechanical Engineers is due to his foresight and faithfulness. Nor has this Society alone been benefited by his work. He had long been a source of information and inspiration to all who were concerned with the affairs of the engineering profession.

Especially should we remember that the existence of the Engineering Societies Building that has done so much to stimulate and consolidate the engineering fraternity is largely the result of his enthusiasm and diplomacy. As a consistent and energetic promoter of the life within the walls of this building, his work has been preeminent and will ever be remembered.

Mr. Rice was a man of remarkably broad and sympathetic nature. All good works claimed his attention and support and his range of activities was surprisingly great, both in this country and abroad. As a natural result he had a world-wide host of friends and acquaintances. He had indeed a peculiar ability to make friends of all manner of men. Few men in any walk of life have made so many personal contacts as he, and this in no small way redounded to the credit and reputation of the Society. The loss of these many connections can never be regained.

He was particularly considerate of young men and their problems. Many such sought him out for the advice and help that never failed and which was given freely and graciously. His unfailing good humor and his patience, diplomacy, and steadfastness, even when under criticism, fitted him admirably for the great work that he accomplished as secretary of the Society.

The engineering fraternity has lost a distinguished member, the Society has lost a great leader, and many of us have lost a friend and counselor. Because of such men the race makes progress, and the world is the better for his presence. And his work will endure. The labors of no man, who builds so intelligently and industriously as he, are ever lost but become a part of the inheritance of the race. "Though he were dead, yet shall he live." It is for those of us who remain to build upon the work he directed, so long and so well, the greater and more serviceable society that he always visualized, weaving into it not only technical excellence, but also the fine human qualities that so marked his personality.

"Build thee more stately mansion, O my Soul!
  As the swift seasons roll!
  Leave thy low-vaulted past!
Let each new temple, nobler than the last,
Shut thee from heaven with a dome more vast,
  Till thou at length art free,
Leaving thine outgrown shell by life's unresting sea!"

Dr. Calvin W. Rice's Contribution to International Friendliness[1]

By JOHN H. FINLEY[2]

IT has been announced that I am to give a lecture, but what I shall say will not deserve such a formal or didactic description. It will, I hope, the rather be but as a preface to a series of lectures to bear the name of Calvin Rice—lectures devoted to the causes to which he gave himself, soul, mind, and body. It will so be that though dead he will yet speak on through the years and even centuries, in which we cannot doubt that the earth will be under a unified planetary physical control of engineers with Rice's humanizing and coordinating spirit, so far as man is progressively able to master the forces of nature.

Prometheus, the first of engineers, the demi-god as Shelley recounts his alleviation of man's estate,

> brought the fire to man, tortured to his will iron and gold, the slaves and signs of power, found what was hid beneath the mountains and the waves . . . gave science . . . taught the implicated orbits woven of the wide-wandering stars . . . taught to rule the tempest-winged chariots of the ocean and made the Celt to know the Indian . . . Cities were built and through their snow-like columns flowed the warm winds and the azure ether shone and the blue sea and shadowy hills were seen.

But these were only the beginnings of the ministry of the engineer. He was a Titan in the Greek myth, this half-god, half-man who secreted the fire in a rod of fennel and carried it to mortals, as later the modern engineer, whose name meant originally the man of genius, caught the fire of electricity and carried it in a wire to the inhabitants of all the earth—the mythical precursor of the engineer who has equated the unknown from the known by algebraic signs and evoked the seen from the unseen by divinations of intersecting lines and incantations of logarithms.

Prometheus is no longer bound. A Titan he is free to avail of all that his mind can bring forth to light. You remember into what a rapture Shelley rises at last in his "Prometheus Unbound,"

> To suffer woes which Hope thinks infinite;
> To forgive wrongs darker than death or night;
> To defy Power, which seems omnipotent;
> To love, and bear; to hope till Hope creates
> From its own wreck the thing it contemplates;
> Neither to change, nor falter, nor repent;
> This, like thy glory, Titan, is to be
> Good, great and joyous, beautiful and free;
> This is alone Life, Joy, Empire, and Victory.

It was a Promethean labor that Calvin Rice, Engineer, protagonist of engineers, friend of all mankind, performed, yet happily without Promethean suffering or "withering in destined pain." He went in the midst of his joyous planetary employment, conscious of the appreciation and the goodwill of the world which he was leaving.

I have often used the phrase "planetary consciousness" as something deeper than international-mindedness, importing a basic human relationship without regard to race or nationality. I think from what I have heard that few possessed that consciousness more highly developed than Calvin Rice—and not merely as an official acquaintanceship. Dr. A. B. Hart, Professor of Government Emeritus in Harvard University, has sent me a letter which illustrates that particular phase of the life of Calvin Rice:

> I have received the invitation to be present at your address in the Calvin W. Rice Lecture Series on "International Friendliness." Surely that term is applicable to Mr. Rice, who belonged to the large group of international scientific men; for whatever race, condition, language, or government, there is but one world group composed of those who adhere to the principles of that modern religion which is founded on the establishment, recognition, and application of truth. A more candid man than Calvin W. Rice could not be found, and he was a remarkable example of the international spirit of personal goodwill, of recognition of achievements, and of distribution of the results of research and generalization for the common welfare of mankind.
>
> It is that universality of research, that generous distribution of the results of research throughout the world which characterizes the science of the Twentieth Century.
>
> Added to that indelible search for and acceptance of the truth was the personal side of the man—the humor, the good temper, the forecasting, the belief in the universality of research and of learning Calvin W. Rice is a shining example of that true, indivisible, and universal science which is the pride of our age.

Here are particular instances of his "international friendliness" in which his planetary consciousness of the common fate of mankind expressed itself—first from Dr. L. S. Rowe, Director General of the Pan American Union:

> Mr. Rice was a constant and enthusiastic collaborator in the work of the Pan American Union. He was particularly helpful to us in bringing about closer relations between the engineering societies in Latin America and the national engineering societies of the United States. In the furtherance of this work he made a trip through the countries of South America and was warmly received everywhere. In devising plans for the development of closer cultural ties between the United States and Latin America I constantly turned to him for advice and suggestion. In a word, he was one of the most efficient collaborators in the furtherance of the plan for closer understanding between the nations of this continent.

Next a letter from the General Secretary of the Committee on Friendly Relations Among Foreign Students, Mr. Charles D. Hurrey:

> Dr. Calvin Rice was an esteemed member of our Committee on Friendly Relations Among Foreign Students for the past six years. He was never too busy to receive foreign student callers and to advise them with reference to obtaining practical experience in engineering before returning to their native lands. I have never known a more thoughtful man in writing letters of sympathy and appreciation to foreign student leaders; because of his world travels he was literally a citizen of the world and always manifested a profound respect for our visiting students and professors from abroad.
>
> Outstanding in his international services was his support of the plan by which several hundred German graduates of technical schools came to America after the war to obtain practical training in American industry. Dr. Rice was one of the moving spirits in initiating and carrying through this plan, which has established an abiding friendship between American industrial leaders and hundreds of the brightest younger business and professional men of Germany.
>
> I have in my files many letters from Dr. Rice, and in my memory is a happy recollection of many conversations in which he expressed a conviction that an engineer must be first a real, intelligent world citizen and secondarily an efficient technician.

One more only may I read—from Dr. S. P. Duggan, Director of the Institute of International Education:

> Dr. Calvin Rice was of inestimable aid to me in establishing one of the important activities which took place almost at the beginning of this Institute's career; namely, the exchange made with France

[1] Extracts from address delivered on December 5, 1934, during the Annual Meeting, New York, N. Y., of The American Society of Mechanical Engineers.

[2] Associate Editor of the *New York Times*.

of engineering professors. . . . His own efforts in . . . respect (to securing funds) and others in connection with the exchange were of great value to us.

He has since been a real friend to all our undertakings, particularly to young foreign exchange students of engineering who have come to study at American colleges and universities. His advice was eagerly sought by them and we have felt it a privilege to have such an authority place himself at our disposal.

These are only several instances of his friendship; but it is difficult to enumerate all the occasions when he was helpful. It is needless to say that the Institute and myself feel that the passing of Calvin Rice is an irreparable loss.

His passing would be an irreparable loss if his life were not a prophecy of better human relations as well as physical comforts. His humanizing and coordinating spirit will live on.

Memorial Notices

HAROLD FRANCIS ADAMS (1868–1934)

Harold Francis Adams, whose death occurred at Sayville, L. I., N. Y., on January 30, 1934, was born at Tansborough, N. J., on August 3, 1868, the son of Charles Edwin and Ellen Maria (French) Adams. His early education was secured in the schools of Atlantic City, N. J. After completing high school there he studied at Pierce's Business School and the Franklin Art Institute, both in Philadelphia, Pa., and then at the University of Pennsylvania, from which he secured a certificate of proficiency in 1897.

During the next five years Mr. Adams was a practicing architect in Atlantic City. From 1902 to 1904 he was associated with Herbert D. Hale and H. G. Morse, Jr., in charge of their Philadelphia office as chief draftsman and also in charge of the work of designing the interior fittings of the S.S. *Manchuria* and S.S. *Mongolia*, built by the New York Shipbuilding Company, Camden, N. J. During the following year he was employed by Wilson Eyre, of Philadelphia, directing the construction of a large private home in Jenkintown, Pa.

From 1905 to 1907 Mr. Adams was again connected with Hale and Morse, with whom James Gamble Rogers was also associated at that time. Mr. Adams held the position of assistant chief draftsman and had charge of exterior work on a number of large buildings in New York, including the McCreery store and Engineering Societies Building.

In 1907 and 1908 he was employed by the Purdy & Henderson Co., of New York and Havana, in charge of drawings for the Custom House and other buildings in Santiago de Cuba, and from 1917 to 1920 he was chief draftsman and head of the Havana office of that company and made designs for both concrete and structural steel buildings there. During the years 1909–1917 he worked successively for Mr. Rogers, directing construction on the estate of E. S. Harkness, New London, Conn.; for Goldwin Starrett & Van Vleck, New York, on plans for the Albany (N. Y.) high school; and for Willauer, Shape & Bready, New York, as chief draftsman, working on plans for a number of large buildings in that city.

While in Havana with Purdy and Henderson Mr. Adams attracted the attention of the Royal Bank of Canada, and in 1920 he became manager of its Bank Premise Department there. Later he was transferred to Montreal, Canada. He left the bank's employ in 1923 to open an office of his own in New York, where he engaged in architectural work until his retirement in 1931.

Mr. Adams became a member of the A.S.M.E. in 1921 and also belonged to the American Institute of Architects and the American Arbitration Association, as well as to the masonic fraternity. He had served as captain of the Morris Guards of Atlantic City. He invented a honing device, which was patented in 1918, and was interested in photography. He is survived by his widow, Minnie A. (Rocks) Adams, and two sons, Harold Francis, Jr., and Lewis Edwin, all of Sayville, and a daughter, Helen B. Adams, of Baltimore, Md.

MARION EUGENE BABIONE (1908–1934)

Marion Eugene Babione, whose death occurred on June 2, 1934, was born on May 6, 1908, at Luckey, Ohio, the son of Dr. Augustus A. and Anna (Bollini) Babione. He secured an A.B. degree at Adelbert College, Western Reserve University, Cleveland, Ohio, in 1930, and two years later a B.S. degree in mechanical engineering at the Case School of Applied Science. While at Adelbert College he was elected to the Pi Kappa Alpha fraternity, and he later became a member of Sigma Xi, honorary scientific fraternity. He became a junior member of the A.S.M.E. in 1932.

For eighteen months prior to his death Mr. Babione was employed as mechanical engineer for the Hanna Coal Company, Cleveland.

JOSEPH E. BACHELDER (1868–1932)

Joseph E. Bachelder, who died in Chicago, Ill., on December 12, 1932, following two months' illness, was born at Palmyra, Maine, on November 3, 1868. At an early age his family removed to Holyoke, Mass. Following his graduation from high school, he entered the employ of the Deane Steam Pump Company of Holyoke, progressing rapidly in the organization, and from 1897 to 1901 was assistant to the superintendent of the company. He then went to Chicago to become manager of the Pump Sales Department of Fairbanks, Morse & Co., and remained in the employ of this company, with the exception of a year as vice-president and manager of the Temple Pump Company, Chicago (1914–1915) until his retirement in 1930. From 1901 to 1906 he was manager of the Pump Sales Department. He then was transferred to the Pump Manufacturing Department at Beloit, Wis., where he remained until 1914. After the year with the Temple Company he was appointed engineering salesman and New York branch manager of the Fairbanks organization. He was transferred to Baltimore in 1922 and to St. Louis in 1923, serving as manager in each city.

Mr. Bachelder became a junior member of the A.S.M.E. in 1905 and an associate of the Society in 1921. He is survived by his widow, Grace Swift Bachelder, and by a son and two daughters.

HERBERT A. BARRE (1875–1934)

Herbert A. Barre, chief engineer of the Southern California Edison Company, Ltd., died suddenly of a cerebral hemorrhage at his home in Los Angeles, Calif., on June 28, 1934. He had been in the continuous service of the Southern California Edison Company and its predecessor companies for 23 years, holding successively the positions of electrical and mechanical engineer, executive engineer, and chief engineer.

Mr. Barre was born at Pictou, N. S., Canada, on January 26, 1875, the son of John R. and Elizabeth F. (Merriman) Barre. He went to San Francisco while still a child and received his education in the schools of that city. He secured a B.S. degree from the University of California in 1897 and was a member of Tau Beta Pi, honorary engineering fraternity. He was given an honorary degree by the California Institute of Technology in 1924.

Practically all of Mr. Barre's engineering work was done in the West in connection with electric utility systems. Following his graduation in 1897 he secured employment as electric station operator for the Redlands (Calif.) Electric Light & Power Co. and subsequently held similar positions with the Los Angeles Railway and the Mt. Whitney Power Company, Hammond, Calif., and was assistant electrical engineer with the traction company at Oakland Calif. From 1900 to 1902 he was connected with the Independent Electric Light & Power Co. and Independent Gas Company, San Francisco, as inspector of construction and assistant to the chief engineer of the former, and assistant superintendent of the latter. In 1902 he returned to Los Angeles as assistant to R. S. Masson, consulting engineer of the Henry P. Huntington properties, which included the Los Angeles Railway, the Pacific Electric Railway, and the San Joaquin Light & Power Co.

In 1906, Mr. Barre became associated with the Electric Bond & Share Co., of New York, N. Y., where he was engaged in engineering valuation and operating activities. Construction of the Glenwood and Boulder plants of the Denver Gas & Electric Co. took Mr. Barre West again to Denver in 1907. He became interested in the formation of the Electric Construction Company, which built the Fossil Creek plant of the Arizona Power Company, and was actively engaged in the building of that plant.

In 1911, when the Pacific Light & Power Corp. (a Huntington property, merged in 1917 with the Southern California Edison Company) was beginning the construction of the Big Creek-San Joaquin River hydroelectric development with its 240-mile transmission lines to Southern California, Mr. Barre was engaged as electrical and mechanical engineer. It was in connection with the large hydroelectric and steam system of the Southern California Edison Company that his most outstanding achievements were accomplished. He was possessed of high engineering ability coupled with a very sound economic understanding of the utility industry. His recommendations as engineer and economist are permanently written into the great hydroelectric and steam system of the company he served. In 1911, when Mr. Barre became associated with the properties now comprising the Southern California Edison Company, this system consisted of a development of some 180,000 horsepower, represented by a plant investment of about $50,000,000. During his active participation in the development of this system, the plant capacity was enlarged to more than 1,250,000 horsepower and the investment in properties increased to $350,000,000.

Probably the most outstanding achievement of Mr. Barre's work was the successful construction and operation of the first 150,000-volt long-distance electric transmission lines in the world, transmitting energy from the water power development on Big Creek and the San Joaquin River to Los Angeles, a distance of approximately 240 miles. Later, when the construction of additional power development at Big Creek brought the need for greater transmission capacity to Southern California, Mr. Barre met the requirement by reconstructing the 150,000-volt lines for operation at 220,000 volts. These lines so converted were the first commercial lines in the world to ac-

complish electric transmission at this high voltage. These attainments brought to Mr. Barre world recognition as an engineer and attracted attention from outstanding engineers in America and foreign countries who went to Los Angeles to learn the details of construction and operation. This contribution in the field of electric transmission has not only saved many millions of investment in transmission expense on the projects with which he was associated, but has also had its influence on the development of all the large hydro developments in the world that are distant from the market to which the energy is transmitted.

Mr. Barre became a member of the A.S.M.E. in 1912. He also belonged to the American Institute of Electrical Engineers. He married Annie E. McTear, in Los Angeles, in 1904, and is survived by her and by two children, Ben E. and Ruth E. Barre, and also by his father.

WILLIAM HASTINGS BASSETT (1868–1934)

William Hastings Bassett, a pioneer metallurgist in the non-ferrous metal industry, connected since 1902 with The American Brass Company, died at his home in Cheshire, Conn., on July 21, 1934. He was born at New Bedford, Mass., on March 7, 1868, the son of William A. and Almira D. (Mayhew) Bassett. With forebears who were seafaring men and a childhood spent in such an environment as New Bedford, he had a love for the water which always made boating his favorite recreation. Until within a year of his death he spent a month annually at Martha's Vineyard.

Not choosing to follow the sea as a profession, however, he entered the Massachusetts Institute of Technology, from which he was graduated in 1891 with a Bachelor of Science degree. He spent the next five years as chemist and superintendent for the Popes Island Manufacturing Company, in New Bedford, and then taught chemistry at the Swain Free School there until 1900. During the next two years he was chief chemist at the Newark Laboratory of The New Jersey Zinc Company.

In 1902 Mr. Bassett went to Torrington, Conn., as chemist for the Coe Brass Manufacturing Company, a branch of the newly formed American Brass Company of Waterbury. The following year he was made chief chemist and metallurgist of The American Brass Company, in 1912 its technical superintendent and metallurgist, and in 1930 metallurgical manager of the company. Of his position in the non-ferrous metal industry and the achievements which led to it, Dr. H. Foster Bain, formerly secretary of the American Institute of Mining and Metallurgical Engineers, wrote in an appreciation published in *Mining and Metallurgy* for September, 1934, as follows:

"After beginning as chemist he rose through various grades of laboratory and plant service with ever increasing responsibilities until he became metallurgical manager for the entire company, itself the largest producer of brass and copper goods in the world. He was the recognized unofficial dean of metallurgists in his chosen field long before he died.

"To him, more than any other single individual, though many had part in the movement, was due the introduction and development of technical control of production in copper-alloy manufacture. When he went to Torrington, brass making was an art handed down from father to son or master to apprentice. The manager of a plant was in the hands of his melters who kept close in their own control knowledge of composition, temperature, and time necessary to produce given quantities in an alloy. Results were irregular, unexplained, and often unsatisfactory. Bassett being a chemist at once began taking samples and making analyses. Also, as Holland and Pringle have related in a most interesting chapter in 'Explorers in Science,' he had a microscope. It was at least among the first of those instruments put into effective use in the industry. The result was that composition and quality were accurately related and the metallurgist came to give directions for what went into the pots and when. . . . A firm foundation was laid for large-scale manufacture of standard goods even from high-priced metals.

"Mr. Bassett's researches in alloys were numerous and wide ranging. A large number of patents were secured in his name or in that of members of his staff and still others were bought by the company on his recommendations. It was not only in chemical matters that he was an authority but in much besides. An important reinforced hollow copper cable is one of his inventions and his wide knowledge and sound judgment of methods of testing were recognized by his professional compeers in his election to the presidency of the American Society for Testing Materials which he held at the time of his death."

Mr. Bassett was president of the A.I.M.E. in 1930. Prior to that he had served as chairman of the Institute of Metals Division of the Institute and was instrumental in bring about the merging of the older Institute of Metals with the A.I.M.E. He was awarded the James Douglas Medal by the A.I.M.E. in 1925 "for constructive research in copper and brass and other non-ferrous metals and their alloys and for his contributions to the establishment of high standards of quality."

Mr. Bassett's service in the American Society for Testing Materials was also outstanding. C. L. Warwick, secretary-treasurer of the Society, has summarized it in the following paragraph:

"In June, 1934, after serving a two-year term as vice-president, he was elected president of the American Society for Testing Materials, with which he had been affiliated since 1903. As a personal member and the official representative of his company, he participated actively in the work of a number of A.S.T.M. committees. He took a leading part in the organization in 1909 of the first two A.S.T.M. standing committees in the non-ferrous field, B-1 on Copper Wire and B-2 on Non-Ferrous Metals and Alloys, and served continuously on these committees until his death. He served as a member of the society's executive committee in 1916–1918. He always took a keen interest in advancing the knowledge of the properties of non-ferrous metals and supported actively a number of research and standardization projects in the society."

Mr. Bassett also took part in the work of the National Research Council and during the World War was a member of the War Department's Advisory Committee on Materials for Airplane Construction. He had served on the Metallurgical Advisory Committee to the Bureau of Standards and on the Army Ordnance Advisory Committee, Watertown Arsenal. He was a member of the Army Ordnance Association.

Mr. Bassett became a member of the A.S.M.E. in 1924. He was a former director of the American Institute of Chemical Engineers, a Fellow of the American Association for the Advancement of Science, and held membership in many scientific and engineering societies in this country and abroad, including the American Chemical Society, Society of Automotive Engineers, American Electrochemical Society, Mining and Metallurgical Society of America, American Foundrymen's Association, Society of Naval Architects and Marine Engineers, American Welding Society, The Franklin Institute, American Geographical Society, Institute of Metals and Society of Chemical Industry, London, and Deutsche Gesellschaft für Metallkunde. He was a 32d degree Mason and a Shriner, trustee of the public library in Cheshire, and a deacon in the First Congregational Church there since 1914. He belonged to the Engineers' and Chemists' Clubs in New York, as well as to clubs in Waterbury and Torrington.

Mr. Bassett married Sarah Hedge Whiting in 1892 and is survived by her and by two children, Alice Whiting Bassett and William Hastings Bassett, Jr., the latter technical superintendent and metallurgist for the Anaconda Wire & Cable Co., Hastings-on-Hudson, N. Y.

HENRY HOWELL BAUMGARTNER (1893–1933)

Henry Howell Baumgartner died on November 16, 1933, at his home in Belleville, N. J., of injuries sustained in an automobile accident during the summer. He was president and treasurer of the Essex Engineering Company, organized at Belleville in 1928.

Mr. Baumgartner was born at Flemington, N. J., on July 10, 1893, the son of Frederick and Marguerite (Burkitt) Baumgartner. He attended the United States Steam School at Hoboken, N. J., for five months during the winter of 1917–1918 and served in the United States Navy until early in 1920. He then spent about a year as second engineer on the Standard Oil tanker, "S. V. Harkness." He was refrigerating engineer for Swift & Co. from September, 1921, to February, 1922, and spent the remainder of that year in his former position of second engineer of Standard Oil tankers.

From then until June, 1925, Mr. Baumgartner was connected with the Public Service Gas & Electric Co., of New Jersey, as assistant boiler room engineer and technical assistant. During the next three years, until he organized the Essex Engineering Company, he was refractory and combustion engineer for the Drake Non-Clinkering Furnace Block Company, of New York, N. Y. Several patents, dealing with furnaces and combustion methods, were issued to Mr. Baumgartner.

Mr. Baumgartner became an associate-member of the A.S.M.E. in 1928, and was past-president of the National Association of Power Engineers. He belonged to the Belleville post of the American Legion. His widow, Ida E. (Hughes) Baumgartner, whom he married in 1918, and two sons, William Darrow and Richard Balz Baumgartner, survive him.

JOSEPH ROBERT BLAINE (1884–1934)

Joseph Robert Blaine died at his home in Oak Park, Ill., on December 17, 1934. He had been connected since 1913 with the Miehle Printing Press & Manufacturing Co., Chicago, in charge of the design of offset printing presses.

Mr. Blaine was born in Chicago on November 14, 1884, the son of John Smith and Frances (Birkholz) Blaine. He received a B.S. degree at the University of Wisconsin in 1905, and an M.E. degree from that institution in 1911. For two years prior to entering college he had worked in the shop and toolroom of Edward P. Allis, Milwaukee, and for a year on the erecting floor of Pawling & Harnischfeger, in that city. He spent his vacations in the latter work, also, and after his graduation in 1905 continued with the company for two years as mechanical draftsman.

From 1907 to 1913 Mr. Blaine was engaged by the Berlin Machine Works, Beloit, Wis., as designer of band-sawing machinery, spending four months of the time in Europe in special work for the company. He held some patents on woodworking machinery but his most important work was in the graphic arts field, in which he held about fifty patents for inventions on typographic and lithographic presses and other printing equipment.

Mr. Blaine became a member of the A.S.M.E. in 1917. He is survived by his widow, Mrs. Attolle (Frost) Blaine, a daughter, Virginia P. Blaine, and a son, Robert F. Blaine.

DAVID REES BOWEN (1865–1934)

David Rees Bowen, vice-president of Farrel-Birmingham Company, Inc., Ansonia, Conn., and for many years its chief engineer, died at his home in Ansonia on December 29, 1934. He played an important part in the development of machinery for non-ferrous rolling mills, sugar-cane grinding, stone and ore crushing, and for the manufacture of paper, rubber, linoleum, and other products, and was widely known in the industries where such machinery is used.

Mr. Bowen was born in Cwmavon, Wales, on October 22, 1865. He received his early education in the British national schools and at Llandovery Collegiate School. At the age of 17 he came to the United States with his parents and in 1883 entered the employ of the Farrel Foundry & Machine Co. as a machinist's apprentice. After serving his apprenticeship for three years he worked as a machinist two years longer, then entered the drafting room of the company. His exceptional ability soon led to his appointment as superintendent, then chief engineer. He held the latter position until 1930, when he was elected vice-president, in charge of engineering. In July, 1933, when ill health compelled him to relinquish the active direction of engineering, he became consulting engineer for the company, continuing also as vice-president until his death.

Mr. Bowen had been a member of the A.S.M.E. since 1899. He was also a member of the masonic fraternity and the Elks. He took an interest in civic affairs, serving as alderman of Ansonia and as a member of the board of trustees of the public library there. He was also a vestryman of Christ Episcopal Church. He was very fond of books and through his reading had acquired a fund of knowledge on many subjects. He spoke the Welsh language fluently and was deeply interested in his native land, to which he returned for many visits.

He is survived by his widow, a son, a daughter, and two grandchildren.

HENRY K. BROOKS (1869–1934)

Henry K. Brooks, president of the Capital Iron Works Company and of the Steel Fixture Manufacturing Company, of Topeka, Kan., died in that city on August 12, 1934, of a cerebral hemorrhage.

Mr. Brooks was born at Kettering, Northamptonshire, England, on January 8, 1869, and was educated in that country. His early schooling was received through private instruction and at the Wisbech Schools, Cambridgeshire. After securing government certificates in drawing, he served an apprenticeship with the firm of Gimson & Co., general engineers, of Leicester, at the same time attending the Leicester Technical Evening College. Upon the completion of his apprenticeship he continued in the employ of Gimson & Co. for several years. He also worked for a short time for the Midland Railway Company, in England, before coming to the United States.

During his first years in this country Mr. Brooks was also engaged in railroad work, being employed by the Atchison, Topeka & Santa Fe, the Southern Kansas, and the Southern Pacific railway companies. In 1895 he accepted a position as instructor at the Kansas State Agricultural College, in charge of the foundry, blacksmith, and machine departments. The following year he returned to the industrial field, in the capacity of general manager for the Capital Iron Works. A short time afterward he and his brother, George W. Brooks, purchased the company and he became its president. He spent three years with the company, during this first period, and was responsible for the work under many federal, state, and railroad contracts for structural iron and machinery.

In 1899 Mr. Brooks became interested in the electrical field and took a position with the Electric Axle Light & Power Co., of New York, N. Y., in charge of its equipment in operation on railroads. The following year he became assistant chief engineer for the Consolidated Railway Light & Equipment Co., New York, and for several months had charge of its factories in New York and Connecticut. Later in 1900, having made some improvements in electric train-lighting apparatus, Mr. Brooks was offered a position as chief engineer of the United States Light & Heating Co., New York. He remained with this company until 1908 and was in complete charge of its factory and all mechanical apparatus.

In 1908 Mr. Brooks returned to Topeka to again take over the management of the Capital Iron Works, and he and his brother also formed the Steel Fixture Manufacturing Company for the manufacture of steel office furniture. These two fabricating companies are among the largest of their kind in Kansas and their products are widely distributed.

Mr. Brooks became a member of the A.S.M.E. in 1907. He was a charter member of the Kansas Day Club, New York, N. Y., a member of the board of directors of the Kansas State Historical Society, past-president of the Associated Industries of Kansas, past-president of the Lakeview Hunting and Fishing Club of Topeka, and a Scottish Rite Mason.

Mr. Brooks married Miss Edith Harrison, of Ottawa, Kansas, in 1895. She died in 1927. He is survived by a daughter, Mrs. R. E. Kreuger, of Topeka, as well as by a number of brothers and sisters.

JAMES PARK CALDERWOOD (1884–1934)

James Park Calderwood, head of the Department of Mechanical Engineering at Kansas State College of Agriculture and Applied Science, Manhattan, Kan., died at his home there on August 9, 1934. He is survived by his widow, Coral E. (Parker) Calderwood, whom he married in 1910.

Professor Calderwood was born at Middleport, Ohio, on January 1, 1884, the son of Andrew and Christina (Thompson) Calderwood. He was graduated from the Middleport High School in 1903 and from the Pomeroy (Ohio) Academy the following year, and in 1908 secured his M.E. degree at Ohio State University. The following year he held a fellowship in experimental engineering at the university.

In 1909 Professor Calderwood became instructor in mechanical engineering at Pennsylvania State College. He was made assistant professor in charge of all heat-engine subjects the following year and in 1911 took charge of the experimental engineering laboratory. In 1913 he was advanced to an associate professorship, with the administration of the experimental engineering department, and he continued in this post until 1917. In addition to his regular teaching and administrative work he designed and installed a heating and ventilating system for new liberal arts and mining buildings.

Special research work conducted by Professor Calderwood while at Pennsylvania State College included an investigation of the effect of the velocity and humidity of air on heat transmission through building material; a study of the pounding in gas engines when using kerosene as a fuel; the development of a method of determining the ultimate from the proximate analysis of coal; and a study of the purchase of coal by specification, which was primarily a study of the methods of coal sampling. The results of all of these investigations were published by the Engineering Experiment Station of the college.

Professor Calderwood served as a member of the Pennsylvania State Board for the examination of boiler and elevator inspectors from 1910 to 1917, and was offered, but declined, an appointment as mechanical engineer for the Department of Labor and Industry of the State of Pennsylvania. In 1916 the Pennsylvania State College conferred the degree of Master of Science upon him.

In 1917 Professor Calderwood accepted a position in the Engineering and Inspection Division of the Travelers Insurance Company at Hartford, Conn. He spent a year with the company, working on special engineering problems connected with liability and indemnity insurance, including the special inspection of industrial plants, studies on accident prevention, and the development of a method for the rating of chemical risks. During this year he also served as assistant engineer for the State of Connecticut in connection with the United States Fuel Administration.

Professor Calderwood had been a member of the faculty of Kansas State College since 1918, when he became professor of steam and gas engineering there. He was made head of the Department of Mechanical Engineering the following year. Reports on some of his work there have been published by the Engineering Experiment Station of the college. He had also made frequent contributions to the technical press and was co-author of several textbooks, including "Engineering Thermodynamics," in collaboration with James A. Moyer; "Elements of Steam and Gas Power," with A. A. Potter; and "Elements of Engineering Thermodynamics," with Moyer and Potter. He had been called upon by the public service commissions

of Kansas and Missouri and the Federal Court of Nebraska in cases involving controversy over natural gas.

Professor Calderwood became a junior member of the A.S.M.E. in 1913 and a member six years later. He had served two terms on the executive committee of the Mid-Continent Section of the Society, being vice-chairman in 1925–1926 and again a member of the committee in 1929-1930. He was honorary chairman of the Student Branch of the Society at Kansas State College from 1925 to 1932.

He was also a member of the Society for the Promotion of Engineering Education, was president of its Kansas-Nebraska Section in 1930, and a member of its membership committee in 1928. He belonged to the Kansas Engineering Society and served on its legislative committee in 1925, was faculty advisor of Sigma Tau, of which he was a member, and was advisory editor of the *Kansas State Engineer*. He was also a member of the masonic fraternity, Delta Upsilon, and the honorary fraternities, Phi Kappa Phi and Sigma Xi, as well as of a number of clubs.

ADAMS BURTON CHAMBERLAIN (1875–1934)

Adams Burton Chamberlain, assistant superintendent of the Station Engineering Department of The Edison Electric Illuminating Company of Boston, died at his home in Wollaston, Mass., on January 22, 1934.

Mr. Chamberlain was born in LaCrosse, Wis., on April 1, 1875, the son of Eli and Mary L. (Porter) Chamberlain. At an early age he went to Newport, Vt., to live and was graduated from the St. Johnsbury Academy. After that he was successively employed by the Tremont and Suffolk Mills, Lowell, Mass., on cost and inspection work, by C. R. Makepeace, mill engineer, Providence, R. I., as a draftsman, and by the New England Telephone & Telegraph Co. on construction work.

In January, 1901, he entered the employ of The Edison Electric Illuminating Company of Boston in the Station Engineering Department, where he rose through various grades to the position he held at the time of his death.

Mr. Chamberlain became a member of the A.S.M.E. in 1911. He was also a member of the Engineers Club of Boston and a 32d degree Mason. He is survived by his widow, L. Josephine Eaton, whom he married in 1903, and by a daughter, Kathryn.

WALTON CLARK (1856–1934)

Walton Clark, consulting engineer and former vice-president of the United Gas Improvement Company, Philadelphia, Pa., died in that city on July 30, 1934. He was the inventor of a process for the complete gasification of coal, and contributed to the development of processes for operating water-gas sets. He was the author of numerous scientific articles.

From 1907 to 1924 he was president of The Franklin Institute and in 1926 the institute honored him with its first award of the Walton Clark Medal, for his "distinguished and outstanding contributions" to the gas industry. At the time of the presentation of the award Dr. W. C. L. Eglin, president of the institute, said that Mr. Clark was "a pioneer in the association of physical and chemical laboratories with the development work of gas companies."

Mr. Clark was born in Utica, N. Y., on April 15, 1856, the son of Erastus and Frances (Beardsley) Clark. He attended the public schools of Utica until his seventeenth year, when he entered the service of the New Orleans Gas Company, where he served in various capacities until 1886. In 1883 he was temporarily assigned to the United Gas Improvement Company to complete the installation of a water-gas plant in New Orleans, work on which had been interrupted by a yellow fever epidemic. Mr. Clark was immune to the disease, and so was able to finish the installation, which would have been impossible for an unacclimated engineer.

He served successively in gas companies in Jefferson City, La., and in Kansas City, and following the death of the assistant general superintendent of the United Gas Improvement Company, Mr. Clark was appointed, in August, 1888, to take his place. Six years later he was made general superintendent, a position which covered the duties of the chief engineer of the company and supervisor of the many enterprises under its control. He became a vice-president of the company in 1904 and held that office until he resigned from active duties in 1923. He had been consulting engineer to the company since then.

In 1904 the honorary degree of Master of Engineering was conferred upon Mr. Clark by Stevens Institute of Technology, and in 1911 he received the degree of Doctor of Science from the University of Pennsylvania.

He was past-president (1906) of the American Gas Institute, of which he was one of the founders; became a member of the A.S.M.E. in 1890; and also belonged to the American Institute of Electrical Engineers and the American Society of Mining and Metallurgical Engineers. He was chairman of the board of trustees of the Free Correspondence School for Gas Works Employees.

He also was a member of the Society of the Cincinnati, the Society of the Founders and Patriots of America, and the Sigma Psi fraternity. His clubs included the Engineers' Club, New York, and in Philadelphia the Engineers, Union League, University, and a number of country and athletic clubs. His home was at Chestnut Hill, Pa.

Mr. Clark in 1880 married Miss Alice M. Shaw, of Natchez, Miss., who died in 1882. In 1885 he married Miss Louise Beauvais, of New Orleans. Surviving him, in addition to Mrs. Clark, are Frank Shaw Clark, a son by his first marriage; and three sons and one daughter by his second marriage, Walton, Jr., Theobald Forstall, Beauvais, and Darthela Clark.

WILLIAM JAY COFFIN (1875–1934)

William Jay Coffin, connected with the New York State Department of Public Works since 1922, died at Wethersfield, Connecticut, on June 14, 1934, after several months' illness.

Mr. Coffin was born in Albany, N. Y., on December 20, 1875, the son of William Latham and Anna McHarg Coffin. He received his M.E.E. degree from Cornell University in 1898 and immediately began work as electrical repairman in the Signal Department of the New York Central Railroad, Albany. He was connected with this railroad, in various capacities, until 1914. He served an apprenticeship in the Motive Power Department in 1900 and 1901 and worked in that department successively as stationary engineer, roundhouse foreman, and mechanical foreman at Albany until 1909. He was then sent to the general manager's office in New York, N. Y., to take charge of locomotive tests and tonnage rating. After several months in that work he was made assistant to the superintendent of motive power of the Western Division, investigating mechanical problems in shops and on the road. In April, 1910, he was appointed assistant engineer in the president's office in New York, where he remained until 1914, making technical investigations dealing with locomotive operation and tonnage rating for the entire New York Central system.

In January, 1914, Mr. Coffin was appointed first deputy commissioner of Public Works and he served the City of Albany in that capacity until January, 1922, having charge of the adaptation of motor apparatus to municipal work and establishing repair facilities. He entered the employ of the New York State Department of Public Works as Fleet Commander in March, 1922, in charge of floating equipment on the state barge canal, including tugs and dredges; he also supervised dredging work and wrecking operations. He was made assistant civil engineer in February, 1923, and continued in that capacity until his death. He was assigned to the division of highways of District No. 1, and had charge of several highway construction projects.

Mr. Coffin became a member of the A.S.M.E. in 1923. He also belonged to the masonic fraternity. He is survived by a sister, Helen Coffin, of Hartford, Conn., and by a daughter, Lois Lilian Coffin, of Brooklyn, N. Y. His wife, Selora (Gaskill) Coffin, died in 1922.

SAMUEL DUNLAP COLLETT (1868–1933)

Samuel Dunlap Collett, whose death occurred on December 26, 1933, was born on October 25, 1868, at Newport, Ind. He attended the public schools there and secured his B.S. and M.S. degrees at Rose Polytechnic Institute, where he studied mechanical, electrical, and civil engineering.

Before completing his college work Mr. Collett worked for a short period as a surveyor on the Chicago, Evansville & Chattanooga Ry., took a training course in the factory of the Thomson-Houston Electric Company, and worked in the Boston and Pittsburgh offices of that company.

In the fall of 1895 Mr. Collett entered the employ of the Metropolitan Telephone & Telegraph Co. (now the New York Telephone Company) as engineer in the construction department and for two years was in charge of the interior block department and underground cable work. He then took the position of eastern manager of the Elevator Supply & Repair Co., New York, of which he later became vice-president. He was associated with the company until 1920, and from then until his retirement in 1925 was with the Elevator Supplies Co., Inc., Hoboken, N. J.

Mr. Collett had been a member of the A.S.M.E. since 1902. He was also a member of the American Institute of Electrical Engineers.

ERNEST HARRY CORNELIUS (1890–1934)

Ernest Harry Cornelius, president of the Oklahoma Steel Castings Company, Tulsa, Okla., died in Cleveland, Ohio, on May 8, 1934,

from complications following an operation for appendicitis. The son of Harry B. and Georgiana (Phillips) Cornelius, he was born on October 28, 1890, at Hastings, Neb., and received his early education there. He entered Northwestern University, at Evanston, Ill., at the age of 17, but because of an injury sustained in a fall during his junior year, he was obliged to leave college. After spending about a year and a half in California, he returned to Nebraska and entered the state university at Lincoln, from which he was graduated with a B.S. degree in 1913.

Following his graduation Mr. Cornelius engaged for a time in sales work for the Jacques Steel Company, in northern Texas. During the first part of 1914 he assisted in appraisal work in Kansas City and then took a position with the Metropolitan Street Railway in that city inspecting construction of new trackage and rehabilitation of existing lines. He also made special designs for cars and did other drafting work, remaining with the company for about two years. He spent part of 1916 with the Kansas City Southern Railway Company, as assistant to the bridge engineer. He then moved to Tulsa, and during the next two years was connected with the Oklahoma Structural Steel Company, subsidiary of the Oklahoma Iron Works, as chief engineer and sales manager the greater part of the time.

In January, 1918, Mr. Cornelius began private practice in the design and construction of steel structures, chiefly for the oil industry. This business was incorporated as the Industrial Construction Company two years later, with Mr. Cornelius as president. In 1922 he also organized the Oklahoma Steel Castings Company, which he had since served as president, greatly developing the size of the plant and extending the territory it serves.

Mr. Cornelius took a keen interest in civic affairs and the development of industry in Oklahoma. He was a past-president and for some years had been a director of the Associated Industries of Oklahoma. He had also served as president of the Public Welfare Society of Tulsa, director of the Tulsa Community Fund and the Tulsa Chamber of Commerce, member of the Board of Trustees of the University of Tulsa, president of the Rotary Club in Tulsa, and Member of the Board of Trustees and chairman of the Financial Committee of the Boston Avenue Methodist Episcopal Church. Just previous to his illness he had been appointed to the NRA Code Authority for the steel-castings industry.

Mr. Cornelius became a member of the A.S.M.E. in 1927 and was a former director of the American Iron and Steel Institute. He belonged to Sigma Nu and the masonic fraternities, and to the Tulsa Club and Tulsa Country Club.

In addition to all these activities he found time to take up flying, first as a hobby and later as a means of transportation for business trips.

Mr. Cornelius is survived by three children, Marjorie Ann and Ernest H. Cornelius, Jr., and Mrs. Virginia (Cornelius) Eby. His wife, Virginia Moseley, whom he married in Lincoln, Neb., in 1914, died in 1923.

ALBERT NELSON CRAMER (1883–1934)

Albert Nelson Cramer, whose death occurred on June 21, 1934, was born in Philadelphia, Pa., on March 27, 1883, the son of Harry Howell and Fannie H. (Slack) Cramer. After attending the Manual Training School in Philadelphia for three years, he entered the employ of the Baldwin Locomotive Works there, and for five years worked on detail drawings for the company. He subsequently spent a year and a half each in similar work for the Link-Belt Engineering Company, Nicetown, Pa., and W. E. Hamilton, Columbus, Ohio, and about six months with the Schoen Steel Wheel Company, Philadelphia. In 1908 and 1909 he was engaged in field engineering for P. A. Kley, Philadelphia, and layout work for the Jeffrey Manufacturing Company, Columbus.

During the next ten years Mr. Cramer was designer in charge of experimental development and construction for the Federal Glass Company, Columbus. He then went to Toledo, Ohio, to take the position of assistant engineer, in charge of special engineering development, with the Owens Bottle Company. He left there in the spring of 1929 and spent the next two years as assistant engineer in the research department of the American Optical Company, Southbridge, Mass.

Since July, 1931, Mr. Cramer had devoted all his time to developing inventions of his own. Prior to that date he had already taken out more than twenty patents on glass-making machinery, including blowing, molding, and cutting machines, feeders, and handling apparatus. Subsequently he developed additional equipment for the glass industry, including a number of machines for forming tumblers and hollow glass articles, patents on some of which were pending at the time of his death.

Mr. Cramer became a member of the A.S.M.E. in 1929 and was active in the masonic fraternity in both Columbus and Toledo. He is survived by his widow, Mrs. Alice P. (Bietsch) Cramer, and by a daughter, Elizabeth B. Cramer.

RALPH CROOKER (1854–1934)

Ralph Crooker, a member of the A.S.M.E. since 1890, died at his home in Acton, Mass., on June 2, 1934. He is survived by his widow, Louise C. (Little) Crooker, whom he married in 1905.

Ralph Crooker (3d) was born in South Boston on December 25, 1854, the son of Ralph and Ann A. (Bailey) Crooker. He was graduated from the English High School of Boston with the class of 1872, and had always kept in touch with the members of that class through its annual reunion dinners in Boston.

At the age of eighteen he entered training as a draftsman at the Bay State Iron Works, Boston, under the direction of his grandfather, who was a member of the firm and superintendent of the works. He remained with that company until 1879, when he went to Springfield, Ill., where he spent about six months with the Springfield Iron Company. From 1880 to 1884 he served as engineer and superintendent with the Colorado Coal & Iron Co.

In 1885 Mr. Crooker entered the employ of the Jones & Laughlin Steel Co., Pittsburgh, Pa., and between then and 1907 he spent the greater part of his time with that company, the last period being from about 1903 to the latter part of 1907. His other connections were with the Bessemer Works of the Springfield Iron Company in 1887 and 1888, with the Johnson Company, Lorain, Ohio, during part of 1894, and with the Lorain Steel Company, about 1900.

He had made his home in Acton since 1907 and had not been active in business except for a little consulting work and a few months in 1919 making alterations in piping for T.N.T. for Mackintosh-Hemphill & Co., Pittsburgh. His chief pastime during his latter years was the study of maps and of plans for ocean liners and boats. He also liked to study the rock formations and wild life in the woods near his home.

Mr. Crooker was a Legion of Honor member of the American Institute of Mining and Metallurgical Engineers and also belonged to the Engineers' Society of Western Pennsylvania.

FRANK GARFIELD CUTLER (1881–1934)

Frank Garfield Cutler, of Birmingham, Ala., died on June 17, 1934, at the Manhattan Eye, Ear and Throat Hospital, New York, N. Y., of complications following a mastoid operation.

Mr. Cutler was born on May 13, 1881, at Fort Scott, Kan., the son of Samuel and Ella B. (Dickerson) Cutler. He studied manual training in Louisville, Ky., and then entered the University of Kentucky, from which he received his Bachelor's degree in mechanical engineering in 1901, and the degree of Mechanical Engineer three years later. He was graduated with highest honors and in 1902, when the honorary engineering fraternity, Tau Beta Pi, was organized there, he was made a member.

Immediately upon leaving college in 1901 Mr. Cutler went to Chicago to become assistant in the steam engineering department at the South Works of the Illinois Steel Company. He remained in that position until the fall of 1906, then took charge of the steam engineering department of the Ensley Division of the Tennessee Coal, Iron & Railroad Co. He was made chief of the Bureau of Steam Engineering of the company in 1914 and held that position until his death.

Mr. Cutler was considered an authority on steam power-plant design and construction and especially its application to steel mills. In 1928 he completed studies for and wrote a paper on "Combination Firing of Blast-Furnace Gas and Pulverized Coal," which was presented at the Semi-Annual Meeting of the A.S.M.E. in Pittsburgh in May of that year.[1] These studies were carried on as new boilers and equipment were put in at the mill, and continued practically up to his death.

Mr. Cutler became a member of the A.S.M.E. in 1914 and had served several years on the Executive Committee of the Birmingham Section of the Society, being chairman in 1921. He was secretary in 1917–1918 and president in 1921 of the Alabama Technical Association, and in 1922 was vice-chairman of the Association of Iron and Steel Electrical Engineers. He had also served as chairman of the Jefferson County Orphans Home Association and as a director of the Ensley Rotary Club and a member of the board of governors of the Woodward Gold Club. His hobbies were golf and contract bridge, and especially long automobile trips in both the North and South. He is survived by his widow, the former Blanche B. Duffy, of Chicago, whom he married in 1909, and by three children, Mary E., Samuel M., and Grace L. Cutler.

[1] Trans. A.S.M.E., vol. 49–50, 1927–1928, FSP-50-35, pp. 102–106.

ORTON GOODWIN DALE (1870–1934)

Orton Goodwin Dale was drowned in Barnegat Bay on February 18, 1934, when ice broke beneath him as he was walking on the bay near his summer home at Mantoloking, N. J. Mr. Dale was consulting engineer to the Texas Gulf Sulphur Company, of New York, and a director of the New York & New Jersey Steamboat Company. He is survived by his widow, the former Miss Amy Slade, of Trenton, N. J., whom he married in 1896, and by a daughter, Mrs. D. R. Abbes, of Burlingame, Calif., and two sons, F. Slade Dale, yachtsman, Bay Head, N. J., and Orton G. Dale, Jr., vice-president of Bowne & Co., financial printers, New York.

Mr. Dale was born in Helensburgh, Scotland, on November 8, 1870, but had lived in the United States since he was ten years old. He was graduated from the Stevens Institute of Technology in 1893 and spent the following year in the drawing room of the National Sugar Refining Company, Yonkers, N. Y. He then became New York manager of the Mead-Morrison Manufacturing Company. In 1918 he was connected with the Singer Manufacturing Company, Elizabeth, N. J., and for the next two years was assistant chief engineer of the American International Shipbuilding Corporation, Philadelphia, Pa. He had been with the Texas Gulf Sulphur Company since that time.

Mr. Dale resided in Plainfield, N. J., for some years, but moved to New York in 1916. It was his custom to spend the weekends at his Mantoloking home and he had recently been elected to the Mantoloking Borough Council. He became a junior member of the A.S.M.E. in 1894 and was promoted to the grade of member in 1909. He was appointed a member of the Committee on Publications of the Society in 1922 and served as its chairman in 1925 and vice-chairman the following year.

CYRUS EATON DAVISON (1890–1934)

Cyrus Eaton Davison, formerly chief research engineer for Martin Motors, Inc., New York, N. Y., died on March 8, 1934, after a year's illness. He is survived by his widow, Anna R. (Tong) Davison, whom he married in 1918, and by a son, Robert C., and daughter, Ruth B. Davison, residing at Stapleton, S. I., N. Y.

Mr. Davison was born at Huntsport, Nova Scotia, Canada, on August 26, 1890, the son of Cyrus C. and Laura (Eaton) Davison. He attended the Huntsport High School and at the age of twenty-one entered the employ of the Southern Pacific Company. He served as assistant engineer and chief engineer on various ships of that company until December, 1917. He then entered the United States Navy as an engineer officer, with the grade of lieutenant, serving, during the next year and a half, aboard the U.S.S. *Lakewood* and U.S.S. *Montclair*. During the summer of 1919 he was connected with the United States Shipping Board in the capacity of assistant machinery inspector and chief machinery inspector, supervising the installation of boilers, turbines, and reciprocating engines, and all other mechanical equipment on ships being built by the Federal Shipbuilding Company for the Emergency Fleet Corporation. He was graduated from the Naval Turbine School at Carnegie Institute of Technology, Pittsburgh, Pa., in 1919.

In the fall of 1919 Mr. Davison took charge of the Educational Department of the Ocean Association of Marine Engineers, of which he was a member, and was instructor of its engineering classes for three years. During the following year he was assistant engineer for the French Company, directing the operation and repair of turbogenerators and electric motors.

Mr. Davison again became an instructor in marine engineering in July, 1923, when he took the position of chief engineer and senior engineering instructor at the New York State Nautical School, New York, N. Y. He continued in that work until January, 1926, then spent two years as assistant superintendent of buildings and grounds at Columbia University. He had been with Martin Motors, Inc., since that time, located at the research laboratory in East Rutherford, N. J., at first. He was engaged in the design and construction of internal-combustion engines for test purposes, and in running power and efficiency tests.

Mr. Davison became an associate-member of the A.S.M.E. in 1926 and a member three years later. He also belonged to the masonic fraternity.

WILLIAM EDWARD DEAN, JR. (1888–1933)

William Edward Dean, Jr., who died of pneumonia at Hamilton Ontario, Canada, on December 11, 1933, was a native of Minneapolis Minn., where he was born on June 12, 1888. His parents were William Edward and Elmina (Clapp) Dean. He attended grammar and high school at Los Angeles, Calif., and then served a four-year apprenticeship with the Southern Pacific Company, of that city. In 1910 he entered the University of California, from which he was graduated with a B.S. degree in mechanical engineering in 1914. During his summer vacations he did drafting-room and testing-laboratory work.

Immediately following his graduation Mr. Dean took a position in the Engineering Department of the Westinghouse Air Brake Company, Wilmerding, Pa., and in the fall of that year was made personal assistant to the manager of engineering, to work on the design, manufacture, and performance of air brakes and related devices, such as automatic train control.

In August, 1917, Mr. Dean entered the Second Officers Training Camp at Fort Oglethorpe, Ga. He was commissioned second lieutenant in the Signal Reserve Corps, Aviation Section, and served five months at Park Field, Tenn., and five at Langley Field, Va., as engineer officer in charge of installation and operation of machine, motor-repair, and blacksmith shops. He was then promoted to first lieutenant in the Air Service and until January, 1919, was stationed at Washington, D. C., in the engine and plane maintenance branch of the service.

Upon his return to civilian life, Mr. Dean was made assistant engineer of tests by the Westinghouse Air Brake Company, continuing to carry on research on air brakes. Early in 1921 he was sent to Kingston, Jamaica, on special consulting work for the Jamaica Government Railways. Upon his return he was appointed engineer of tests, and at the beginning of 1926 assistant chief engineer in charge of tests and inspection.

At the close of 1928 Mr. Dean was appointed chief engineer of the Westinghouse Brake Company of Australasia Ltd., and for several years was located in New South Wales, Australia. In August, 1932, he was transferred to the position of chief air-brake engineer of the Canadian Westinghouse Co. Ltd., at Hamilton, Canada, the position he held at the time of his death.

Mr. Dean became a junior member of the A.S.M.E. in 1921 and a member four years later. He also belonged to the Air Brake Association and to the honorary fraternities, Tau Beta Pi and Sigma Xi. He had patented many air-brake system devices.

Surviving Mr. Dean are his widow, Doris (Brown) Dean, whom he married in 1923, and three sons, Frederick Brown, William Corner, and Robert Stanley Dean.

GEORGE PASS DeHAVEN (1886–1934)

George Pass DeHaven, son of William Henry and Margaret M. (Pass) DeHaven, both descendants of old Pennsylvania families, was born at Harrisburg, Pa., on September 3, 1886. His early education was secured in the schools of that city. After his graduation from high school there he studied for two years at the Stevens Institute of Technology, Hoboken, N. J., and two years at Drexel Institute, in Philadelphia. He then entered the employ of the Pennsylvania Steel Company, at Steelton, where he spent a year each in the pattern and machine shops, and about six months each in the iron and steel foundries.

In January, 1914, he took a position in the engineering department of the Bethlehem Steel Corporation, Steelton, where he remained for four years, working on the design of rolling mill machinery. From then until the summer of 1923 he was connected with the Harrisburg Foundry and Machine Works, spending four years on the design, testing, and preparing specifications and estimates for steam engines, then serving successively as assistant purchasing agent, assistant works manager, and production manager.

He was next employed for about a year as designer and checker for the Koppers Company, Pittsburgh, working on gas producers and machinery for operating them, and during the winter of 1924–1925 designed rolling-mill machinery for the Birdsboro Steel Foundry & Machine Co., Birdsboro, Pa. He then became assistant chief engineer for the Taylor-Wilson Manufacturing Company, McKees Rocks, Pa., with which he remained until he was appointed chief engineer, General and Maintenance Employees, of the Bureau of Public Grounds and Buildings of the Commonwealth of Pennsylvania. He subsequently was made superintendent of the Bureau of Maintenance of Public Grounds and Buildings, the position he held at the time of his death, which occurred on February 19, 1934, at his home in Harrisburg.

Mr. DeHaven became an associate-member of the A.S.M.E. in 1926 and in that year he also took some courses in the shaping of steel and roll pass design at the Carnegie Institute of Technology. He had served as corporal and quartermaster sergeant on the Governor's Troop of the Pennsylvania National Guard and belonged to the masonic fraternity and the Elks. He is survived by his widow, Margaret M. (Edwards) DeHaven, whom he married in 1906, and by their son, Charles A. DeHaven.

EDWARD H. J. DILLON (1887–1934)

Edward H. J. Dillon, who died in Philadelphia, Pa., on March 28, 1934, was born in Brooklyn, N. Y., on April 15, 1887, the son of John and Anna (Mohan) Dillon. He attended the public and high schools there, and subsequently studied for two years at Brooklyn Polytechnic Institute and St. John's University, Brooklyn.

At the age of seventeen Mr. Dillon entered the employ of the Griscom-Russell Company, of Philadelphia, marine and stationary power plant engineers and manufacturers, and had spent his entire professional life with that company, rising to the position of manager of the Philadelphia office.

Mr. Dillon became an associate of the A.S.M.E. in 1920 and belonged to the Knights of Columbus. He is survived by his widow, Margaret Dillon, whom he married in 1915.

FRED DOEPKE (1862–1934)

Fred Doepke, founder and president of the Wrought Washer Manufacturing Company, Milwaukee, Wis., died in Los Angeles, Calif., on April 23, 1934, after having been in failing health for several years. Mr. Doepke became a member of the A.S.M.E. in 1922 and since 1926 had served as secretary of the Sectional Committee on the Standardization of Plain and Lock Washers, for which the A.S.M.E. and the Society of Automotive Engineers are joint sponsors. He was also a past vice-president of the Engineering Society of Milwaukee and belonged to a number of social clubs in Milwaukee and vicinity.

Mr. Doepke was born at Rockford, Ill., on March 28, 1862, the son of Gotlieb and Bertha (Smith) Doepke. The family later moved to Milwaukee and he attended school there from 1879 to 1883, at the same time serving an apprenticeship in the machine shops of the old Filer & Stowell Co. After working for a time in 1883 in machine shops in Cleveland, Ohio, he went to New York, where he studied at Cooper Union nights and was employed by E. W. Bliss Co., the North River Iron Works, and the Freeland Tool Works. Subsequently he went to Philadelphia, where he likewise studied evenings, taking courses offered by The Franklin Institute, and worked in the daytime.

During these years Mr. Doepke became interested in automatic machinery and conceived the idea of automatic washer-making machinery. He returned to Milwaukee in 1885 and started in business with A. J. Read. The Wrought Washer Manufacturing Company was founded about 1887 and he served as vice-president of the company until 1910, when he became its president. He not only designed and built the plant itself, but also practically all of its special machinery. Under his guidance the plant became one of the largest in the world devoted to the production of washers.

Mr. Doepke had traveled extensively, visiting many foreign countries, and was widely known in engineering circles and throughout the metal industries. For some years prior to his death he had made his home in St. Petersburg, Fla. He is survived by his widow, Anna M. (Disch) Doepke, whom he married in 1892, and by one son, Fred C. Doepke, of Milwaukee.

ROSCOE IRVING DUNTEN (1888–1934)

Roscoe Irving Dunten, whose death occurred on June 17, 1934, was born at Calcium, N. Y., on April 13, 1888, the son of Milton Thomas and Anna Elizabeth Dunten. To supplement his high-school education he took courses in steam and electrical engineering through the International Correspondence School, completing these studies in 1907. He then secured work as assistant at the steam power plant of the St. Regis Paper Co. at Deferiet, N. Y., where he remained for two years, part of the time being chief engineer.

From 1909 to 1912 Mr. Dunten was in charge of the steam-electric power plant of the Delaware & Hudson R. R. Co. at Oneonta, N. Y., and during the next year directed the operation and maintenance of an 80-ton ice-making plant of the Tampa Ice Company, at Palmetto, Fla.

In 1913 the Southern Utilities Company, of Palatka, Fla., engaged Mr. Dunten as engineer in its electric plant at Bradentown, Fla. After three years there he was transferred to the combined ice and electric plant at West Palm Beach as chief engineer. Something over two years later he was made supervising engineer of all of the company's properties, consisting of more than twenty plants located at different points in Florida and Georgia. He remained with the company until the latter part of 1921, his work including not only the supervision of existing plants but also the design, layout, and erection of new plants and research on burning oil fuel under power boilers.

During the next seven years Mr. Dunten was vice-president and general manager of the San Juan Ice and Refrigeration Company, in Porto Rico. He directed the reconstruction and enlargement of the company's plant at San Juan, greatly increasing the output and reducing fuel consumption. He returned to Florida in 1928 to become manager of the Caribbean Division of Pan-American Airways, Inc., with headquarters at Miami, and held that position at the time of his death.

Mr. Dunten joined the A.S.M.E. with the grade of associate-member in 1921 and was promoted to full membership three years later. He was Rear Commodore of the Biscayne Yacht Club and a member of the masonic fraternity. He is survived by his widow, Luvenia M. (Wood) Dunten, whom he married in 1909, and by a daughter, Florence A. (Dunten) Foster.

GEORGE C. FARKELL (1872–1934)

George C. Farkell, superintendent of the rail and blooming mills, at the Lorain, Ohio, works of the National Tube Company, died at St. Joseph's Hospital in that city of pneumonia on October 22, 1934. He had been connected with the Lorain Works since early in 1905, when he became assistant superintendent of the skelp mills. The following year he was advanced to the position of superintendent of the skelp mills, and in 1907 was made superintendent of the rail and mechanical departments of the works. He held this position for ten years, then was superintendent of the rolling mills until April, 1934, when he was further promoted to the position he held at the time of his death.

Mr. Farkell was born on January 12, 1872, at Canajoharie, Montgomery County, N. Y., the son of John and Catherine Farkell. He was graduated in 1887 from the local high school and entered Cornell University from which he was graduated in 1892 with an M.E. degree in electrical engineering. While at Cornell he was elected to membership in the honorary scientific fraternity, Sigma Xi.

For two years after leaving college Mr. Farkell was engaged in electrical construction work for H. Ward Leonard & Co., New York, and from then until the end of 1895 was on the staff of the Physical Testing Laboratory of the Lukens Iron & Steel Co., Coatesville, Pa. During the next five years he was assistant engineer with the United States Revenue Cutter Service, for three years of that period assistant to the engineer-in-chief detailed for inspection duty and for engineering work in connection with the construction of steam machinery for new revenue cutters.

Mr. Farkell became assistant chief inspector of the Homestead Steel Works of the Carnegie Steel Company in January, 1901. In October of that year he was made assistant to the superintendent of the 128-in. plate mill, the 42-in. universal, and the 30-in. slabbing mill of the Homestead works, where he remained until May, 1904. Between then and his connection with the Lorain Works, he held the position of assistant superintendent of the slabbing, plate, rail, and structural mills of the Lackawanna Steel Company, Buffalo, N. Y.

Mr. Farkell contributed greatly to the development of rolling-mill practice and held patents on a manipulator used in handling rolled-metal shapes; a speed-controlling apparatus for electrically driven rolling mills (joint patent with Edwin S. Lammers); a method of rolling T-shaped rails; and a device for tightening nuts on screw-threaded ends of bolts.

Mr. Farkell had been a member of the A.S.M.E. since 1913. He also belonged to the American Society for the Advancement of Science, the American Iron and Steel Institute, and the masonic fraternity. He was very active in community affairs in Lorain. He was one of the founders of the local Red Cross chapter and served on its executive committee from 1917 to the time of his death, part of the time as chairman. He had been a member of the board of directors of the Lorain Y.M.C.A. since 1925 and was chairman of the committee which had charge of erecting a new Y.M.C.A. building. He was a member of the board of directors of the Lorain Community Fund during the last five years of his life. In recognition of these services he was selected in 1933 for the *Lorain Journal* Achievement Award, as the citizen who during the year had performed the most noteworthy service to the community.

Radio construction was his special hobby and golf his favorite pastime.

His widow, the former Miss Ouida E. Pease, whom he married in 1909, survives him.

THOMAS FARMER (1852–1934)

Thomas Farmer, a member of the A.S.M.E. since 1891, died of pneumonia at the United Hospital in Portchester, N. Y., on November 6, 1934. He is survived by his widow, formerly Miss Maud H. White, whom he married in 1879, and by a daughter, Elizabeth G. (Farmer) Kelsey, and a son, Thomas, the fourth in line to bear that name.

Mr. Farmer was born in Boston, Mass., on October 12, 1852, his

mother being Henrietta B. (Cobb) Farmer. After his graduation from high school in Roxbury in 1869 he learned the machinist's trade at the shops of A. Leitelt Co. in Grand Rapids, Mich. He spent three and a half years there, then went to Aurora, Ill., where he worked in the locomotive shops of the Chicago, Burlington & Quincy R.R. He next returned East, working for a time on erecting for the Providence Machine Company. During these years he studied drawing and other engineering subjects.

In 1874 he returned to Grand Rapids, where he obtained the position of superintendent of the city water works, which he held for five years, resigning to go into business for himself. Owing to lack of capital his venture was not successful and he went to Chicago, where he was employed for a time as tool saleman. He did not find this work congenial, however, and took a position as general superintendent of the Somersworth Western Manufacturing Company of Bloomington, Ill. During two years spent with this company he built new shops and designed special tools. He resigned that position to become superintendent of the Detroit Radiator Company (which later became a part of the American Radiator Corporation). He directed the laying out and building of a new plant and had charge of the design of new tools and the operation of all departments of the plant.

Mr. Farmer's next connection was with the Russel Wheel & Foundry Co. of Detroit, for which he built a new plant and served as its superintendent for a time. About 1894 he became chief engineer of the Detroit Citizens Railway, with which he remained for nine years. From Detroit he went to Cleveland, Ohio, where he was associated for ten months with the Kuhlman Car Company. He then opened an office for consulting work in the street-railway field. He gave this up to rebuild and electrify the plant of the United States Heater Company (subsequently known as the United States Radiator Corporation) in Detroit, after which he became sales manager of the Detroit office of the Warner & Swasey Co. He spent nine years in that capacity, prior to becoming manager of the Detroit office of Manning, Maxwell & Moore, Inc. He had not been active in business since 1925.

EDWIN E. A. FISHER (1870–1934)

Edwin E. A. Fisher was born in Providence, R. I., on August 31, 1870, the son of James Potter and Caroline Martha (Butts) Fisher. He attended Cooper Union, New York, then entered Cornell University, from which he was graduated in 1891 with an M.E. degree. He specialized in electrical engineering during his senior year and his first position after graduation was with the Field Engineering Company, working on the installation of an electric railway between Paterson and Passaic, N. J. After a year in that connection he went to Lynn, Mass., to take a course at the General Electric Company works there. In 1893–1894 he engaged in electric-railway installation for the Union Railway Company, Providence, R. I., and the following year was instructor in physics and electrical engineering at Lehigh University.

From 1895 to 1912 Mr. Fisher was located the greater part of the time outside the United States. He studied physics, chemistry, and mathematics for a year at the University at Göttingen, Germany, after which he was located in England and Canada. For ten years beginning in 1901 he was connected with the Bureau of Education in the Philippine Islands, as division superintendent of schools, superintendent of the Philippine School of Arts and Trades, and acting assistant director of education in charge of schoolhouse construction in the Philippines, and devised the unit system for schoolhouses there. He next spent a year as assistant commissioner of the interior of Porto Rico.

Since his return from Porto Rico Mr. Fisher had been engaged in technical and statistical research in relation to life insurance. He was in the statistician's department of the Prudential Insurance Company, Newark, N. J., from 1912 to 1921 and since then, until his death on March 3, 1934, had been in the risk classification department of the John Hancock Mutual Life Insurance Company, Boston, Mass.

Mr. Fisher became a member of the A.S.M.E. in 1922 and also belonged to the American Statistical Association, American Association for the Advancement of Science, and Boston Life Underwriters Association. He was vice-president and director of Fisher & Thompson, Inc., New York. In Dover, Mass., where he made his home, he was treasurer of the First Parish of Dover and vice-president of the Dover Historical and Natural History Society. He married Sophie L. Armington, of Providence, in 1907, and is survived by her and by three sons and a daughter.

STANLEY GRISWOLD FLAGG (1860–1934)

Stanley Griswold Flagg, president of Stanley G. Flagg & Co., Philadelpha, Pa., died of a heart attack on March 14, 1934, just a week after the death of his wife, Elizabeth (Windrim) Flagg. He is survived by his son, Stanley Griswold Flagg, 3rd, of Philadelphia, and his daughter, Mrs. Edward S. Nugent-Head, of London.

Mr. Flagg was born in Philadelphia on January 21, 1860. His mother was Adelaide (Gordon) Flagg and his father Stanley Griswold Flagg, founder of the company, which manufactures malleable-iron, cast-iron, and brass fittings, brass valves, and specialties. He learned the business under his father and succeeded him to the presidency of the company.

Mr. Flagg was one of the early advocates of standardization in the valve and fittings industry. As a member of the A.S.M.E., which he joined in 1891 and of which he was a manager for the term 1910–1913, he was appointed a representative on the American Engineering Standards Committee soon after its formation in 1918. He continued to serve on that body, later reorganized as the American Standards Association, until 1930.

For some years prior to the formation of the A.E.S.C. he served on special A.S.M.E. committees on pipe and screw threads. He represented the Society on the Sectional Committee on Standardization and Unification of Screw Threads from the time of its organization in 1921 until 1930, and was also the Society's representative on the Sectional Committee on Pipe Threads. He also represented the Manufacturers Standardization Society of the Valve and Fittings Industry on the Sectional Committee on Pipe Flanges and Fittings.

He was a most active worker on these and other committees to which he was appointed by the various technical organizations to which he belonged, losing no opportunity to cooperate in the sound development of the industry as a whole. He made frequent trips abroad, promoting standardization at international conferences and through his many personal contacts.

Mr. Flagg was a past-president of the American Foundrymen's Association and a member of the American Institute of Mining and Metallurgical Engineers, the American Society for Testing Materials, and the Iron and Steel Institute. He had also served as president of the Pennsylvania Society of Sons of the Revolution, vice-president of the General Society of the Sons of the Revolution, and treasurer of the Society of Colonial Wars in the Commonwealth of Pennsylvania. One of his pastimes was the collecting of rare books.

WILLIAM FRAY (1865–1934)

William Fray, who died at the Bridgeport (Conn.) Hospital, on July 22, 1934, after a two years' illness, was born at Cornwall, England, on July 2, 1865, the son of Samuel and Emma (Sweet) Fray. He came to the United States as a boy and attended the Bridgeport public schools. At the age of sixteen he was apprenticed to the machinist and toolmaking trade and he worked up in that field, studying mechanical drawing, mathematics, and other subjects evenings, until he became assistant foreman at the Farrel Foundry & Machine Co., Waterbury, Conn., in 1890, and foreman two years later.

He resigned in 1900 to take the position of superintendent of the wire mill of the Benedict & Burnham Manufacturing Co., Waterbury, with which he remained four years. He then became production superintendent for John S. Fray & Co., Bridgeport, manufacturers of small tools for mechanics. He was there until 1910 when he became superintendent of the Standard Brass & Copper Tube Co., New London, Conn. After two years he was made secretary of the company, as well as superintendent, and two years later became its president and general manager. He greatly enlarged the plant during his connection with it. After the sale of the business in 1917 Mr. Fray retired from active business, but from time to time acted as a consulting engineer on brass and copper-tube manufacture, serving, among others, the Bridgeport Brass Company, Wheeler Condenser & Engine Co., Detroit Brass & Copper Co., Scovill Manufacturing Company, Dominion Copper Products Company, and Philadelphia Roll & Machine Co.

Mr. Fray became a member of the A.S.M.E. in 1920 and was treasurer of the Bridgeport Section of the Society from 1926 to 1931. He was also a member of the Bridgeport Engineers Club and the masonic fraternity. After his retirement he spent some time in travel. He is survived by his widow, Minnie J. (Steiber) Fray, whom he married in 1883, and by a son, George H. Fray, and daughter, Florence (Fray) Clark.

DAVID WILLIS FRENCH (1860–1934)

David Willis French, retired superintendent of the Hackensack (N. J.) Water Company, died at his home in Englewood, N. J., on August 9, 1934, following a few days' illness. He is survived by his widow, Wilburta F. French, and by a sister, Mrs. Laura E. Libby of Boston. His first wife, Amelia (Eddy) French, whom he married in 1884, died in 1929.

Mr. French was born at North Weymouth, Mass., on March 24, 1860, his father's name being John French. At the age of sixteen he entered upon a three-year apprenticeship in the shops of the Whittier Machine Company, Boston, during which period he also attended night school four evenings a week. The following year he took a special course at the Massachusetts Institute of Technology. In 1881 he became associated with the Henry R. Worthington hydraulic works in South Brooklyn, N. Y., as erecting engineer. In 1882 the firm contracted to install a pump for the Hackensack Water Company at the New Milford plant. Mr. French supervised this installation and upon its completion was retained by the Hackensack Water Company. In 1883 he had charge of installing the plant for the high service system at Weehawken, and subsequently was engaged to operate it for a year under Charles B. Brush, chief engineer and superintendent. He was engineer in charge of the high service system for eight years, then was made deputy assistant superintendent of the company, and after the death of Mr. Brush in 1896 was made superintendent of the company, the position he held until his retirement in 1928.

Some of his major projects were the reconstruction of the New Milford pumping plant, the installation of the present great filter system, construction of the sub-pumping station at North Bergen, and the extension of the water system to supply about sixty towns in Hudson and Bergen counties. In 1900 he recommended to the board of directors of the company the purchase of a subsidiary water company located at Spring Valley, N. Y., and in 1912 he reconstructed this plant to supply ten towns in Rockland County.

In 1913 Mr. French perfected a machine which makes it possible to install a line gate without the interruption of service, a machine now widely used throughout the United States.

Mr. French became a member of the A.S.M.E. in 1916. He was also a member of the American Water Works Association, which he served as president in 1909, was a director of the Hoboken Savings Bank, a member of the Union City Masonic Lodge, member of the board of the First Presbyterian Church, Englewood, director of the New Jersey Utilities Association for twenty years, superintendent of the Grove Church Sunday School for a number of years, member of the Executive committee of the Boy Scouts, North Hudson County Council, and a former member of the Board of Education of Union City.

A. O. FRICK (1852–1934)

A. O. Frick, chairman of the board of the Frick Company, Waynesboro, Pa., and active in the affairs of the firm for more than sixty years, died at a hospital in Baltimore, Md., on April 20, 1934.

Mr. Frick was born on June 16, 1852, at Ridgeville (now Ringgold), Md., where his father, George Frick, a native of Switzerland with a genius for mechanical things, then had his engine shop. The family having moved to Waynesboro in 1861, Mr. Frick's education was completed in the public schools of that town. At the age of fifteen he began work as an apprentice in the Frick shops under his father, later becoming foreman, then draftsman, then mechanical engineer.

In the late 60's and the 70's Mr. Frick, with an engineering insight years ahead of his time, improved the designs for the Frick portable steam engines and boilers. Some of his early drawings for these engines, bearing the date 1869, are still preserved, and illustrate with their fine, accurate lines, the craftsmanship he displayed in all his work.

Between 1876 and the early 80's he developed the Frick steam traction engine, doing the ground work which led through such steps as steering by a team of horses, chain drive from the engine shaft to the gearing, the use of a train of small gears to replace the chain, devices for springing the rear axle, feedwater heaters, etc. He also patented an improved balanced slide valve, some of his early drawings showing the adaptability of this valve to locomotive practice.

In 1882 and 1883 Mr. Frick made the drawings for the original refrigerating machines with which the firm pioneered in this field, and he subsequently assisted in enlarging the line of ice-making equipment. About 1889 he spent a year in Springfield, Mo., building and operating an ice plant. Following this he devoted himself particularly to managing the engineering work of the company. He was made vice-president in 1896 and elected to the presidency of the company in 1904, but continued to serve as consulting engineer on all of its work.

Mr. Frick had been chairman of the board since 1924, and though in recent years he had retired from active work, he always took the keenest interest and pride in the company's progress. When he was past the age of 75 he took over the duties of chief engineer for several months, during his absence. He was the last living member of a group of thirteen men who banded together in 1873 to save the firm from the effects of the panic of that year.

In addition to his connection with the Frick Company, Mr. Frick was a director of the Citizens National Bank & Trust Co., a director of the Chamber of Commerce and of the Beneficial Fund Association of Waynesboro, a member of the board of managers of the Waynesboro Hospital, a member of the Waynesboro Country Club, and charter member of the local masonic lodge. He became a member of the A.S.M.E. in 1885.

He is survived by his widow, the former Miss Margaret Mehaffey, a sister, and two brothers, Amos and Ezra Frick.

ALFRED BROOKS FRY (1860–1933)

Alfred Brooks Fry, Commodore, U.S.N.R. (Retired), died in Coronado, Calif., on December 4, 1933. Mr. Fry entered Government service in 1886 as inspection engineer of public buildings under the Supervising Architect's Office of the Treasury Department and held many important posts in civil and naval service prior to his retirement in 1927.

Mr. Fry was born in New York, N. Y., on March 3, 1860, a son of Major Thomas William G. Fry (U.S.V.) and Frances (Olney) Fry, and a great-grandson of Captain Benjamin Fry, who served in the Continental Army. He attended private schools and Morse's School in New York and studied engineering at Columbia University. He spent several years at sea as an oiler and assistant engineer and also had shop experience with the Corliss Engine Company, Providence, R. I., and at Portland, Me., and Boston, Mass., with the Portland Company and Globe Mill Company. He was captain of the Boston Fire Department from 1882 to 1886.

In civil service in the Treasury Department Mr. Fry was advanced from inspection engineer to chief engineer and supervising chief engineer. He had charge of the construction, engineering operation, and repairs of United States public buildings and their mechanical and electrical plants. Much of his time was spent on work at New York but he also had assignments in Boston, Chicago, Washington, New Orleans, and Galveston and on the Pacific Coast and in connection with quarantine stations and marine hospitals on the Atlantic, Gulf, and Pacific Coasts. From 1916 to 1926 he was consulting engineer for the U. S. Immigration Service, in charge of the construction and maintenance of immigration buildings. He served on a Congressional Commission on the mechanical transmission of mails in 1913.

By order of the President Mr. Fry served as a member of the Board of Consulting Engineers for the Improvement of State Canals in New York from 1904 to 1912, and he also studied and reported on European and Egyptian canal, river, and port improvements in 1909 and 1918. He was an engineer member of the Special Panama Canal Commission in 1921.

Mr. Fry was consulting engineer for the Association for the Protection of the Adirondacks, from 1908 to 1920, and for the Department of Water Supply, Gas, and Electricity of the City of New York, from 1914 to 1917. He was a member of the Committee on Water Storage and Waterways of the Merchants' Association of New York from 1918 to 1926.

In 1892 and 1893 he organized the first Engineer Division of the United States Naval Militia at Boston and New York. He served as engineer lieutenant and engineer lieutenant-commander from 1892 to 1910 and as commander and chief of staff of the Naval Militia, New York, from then until 1923, when he was made commodore. He was retired in 1924 with the rank of rear-admiral.

In the U.S.N.R. he was advanced through the ranks from lieutenant to commodore, retiring in 1927. He was acting chief engineer in the U.S.N. during the Spanish-American War, 1898, and engineer-aide to Rear-Admiral Burd, industrial manager of the Third Naval District and New York Navy Yard, during the World War. He was engineer-observer on the first run of the U.S.S. *Leviathan* in 1917 and also saw duty at sea and at English and French ports. He was consulting marine engineer for the U.S. Army Transport Service in 1922. He was a member of the International Congresses of Navigation.

Since his retirement Mr. Fry had served on the Harbor Committee of the San Diego Chamber of Commerce in 1929 and on the City Council of Coronado in 1929–1930 and was mayor pro-tem of Coronado in 1930–1931, and mayor from then until the time of his death. He served on the employment committee of Southern California in 1931.

Mr. Fry became a member of the A.S.M.E. in 1892. He was also a member of the American Society of Civil Engineers, American Society of Naval Engineers, Society of Constructors of Federal Buildings, of which he had been president. He was a member of a committee of engineers representing the National Engineering Societies at the New York Constitutional Convention in 1915. He belonged to the Society of Colonial Wars, The Society of the Cincinnati of the Providence Plantations, and the Military Order of the Spanish-

American War, and his clubs included the Columbia University, New York, and Army and Navy, Washington. He was an honorary member of the Yachting Department of the New York Athletic Club.

Surviving Mr. Fry are his widow, Emma V. (Sheridan) Fry, whom he married in 1890, and their son, Sheridan Brooks Fry.

GEORGE WARREN FULLER (1868–1934)

George Warren Fuller, who became internationally known as a sanitary engineer and expert on water supply and sewage disposal died at his home in New York, N. Y., on June 15, 1934. He was born at Franklin, Mass., on December 21, 1868, and was the son of George Newell and Harriet (Craig) Fuller. After his graduation from the Massachusetts Institute of Technology in 1890 with an S.B. degree, he studied for about a year at the University of Berlin and in the private office of the engineer of the Berlin water works. He then devoted about nine years to research work in water purification, chiefly at the Lawrence Experiment Station of the Massachusetts State Board of Health, and at Louisville, Ky., and Cincinnati, Ohio, where his work on coagulation and rapid sand filtration laid the basis for American water purification practice.

Since 1899 Mr. Fuller had engaged in consulting work, with his headquarters in New York. From 1901 to 1911 he was in partnership with Rudolph Hering under the firm name of Hering & Fuller. During the next few years he continued practice under his own name. In 1916 he formed a partnership with James R. McClintock, under the firm name of Fuller & McClintock, of which he was senior member at the time of his death.

Mr. Fuller served as consulting engineer for major water works and sewerage improvements in more than 150 large cities in this country and abroad. He had been consultant to the New York Board of Water Supply since 1906. In connection with the development of the Catskill reservoirs and supply systems he made an intensive study of the principal systems in Wales and Scotland which led to the installation of aerating stations below the Ashokan and Kensico reservoirs. He also served as consultant to the Metropolitan Sewage Commission and in 1928 and 1929 reported to the city upon the Wards Island sewage-treatment project, plans for which were subsequently drawn by his firm.

In 1917 and 1918, during the World War, Mr. Fuller was a member of a central committee at Washington having to do with the engineering, planning, and sanitation of various army camps, and he was also consulting engineer to the Construction Division of the Army. Directly after the Armistice he served as a member of the Franco-American Engineering Congress which convened at Paris to consider reconstruction and economic problems in France.

One of the many important posts held by Mr. Fuller since the World War was that of chairman of the Engineering Board of Review for the Sanitary District of Chicago, which was formed in 1924 to consider a new sewage-disposal system in that city. The recommendation of the board to draw sufficient water from Lake Michigan to carry sewage in satisfactory dilution through the Illinois Canal to the Mississippi River was adopted and successfully carried out. In 1934 he was recalled to advise a special board of review in regard to additions to this system.

Mr. Fuller had always taken a keen interest in public-health work. He was consulting engineer to the United States Public Health Service in connection with municipal improvements necessary to public health, contributed largely to the development and adoption of the Standard Methods of Analysis for Water and Sewage sponsored by the American Public Health Association, and as chairman of its Council on Standardization, helped the American Water Works Association in the publication of its Manual of Water Works Practice.

Mr. Fuller was a member and chairman of an engineering advisory committee of the Reconstruction Commission of the State of New York, and advisor to the International Joint Commission on Boundary Waters between the United States and Canada. Among the projects on which the advice of his firm was sought during recent years were water works for Shanghai, China; the Wanaque aqueduct, for the New Jersey Water Policy Commission and the North Jersey District Water Supply Commission; new water supply and purification systems for Kansas City, Mo., Memphis, Tenn., and Wilmington, Del.; sewage-treatment works at New Haven, Conn., and Lexington, Ky.; a comprehensive system of intercepting sewers and sewage treatment for the Hackensack Valley Sewerage Commission of New Jersey involving some fifty communities bordering on the Hackensack River and its tributaries; the collection and treatment of sewage and industrial wastes reaching Narragansett Bay from the city of Providence and neighboring municipalities, a study of which was made for the Metropolitan Sewerage Commission of Rhode Island; and the future water supply of the Minneapolis-St. Paul district, a report on which was completed just prior to his death.

Mr. Fuller's work was not entirely in the hydraulic and sanitary fields, however. He was an expert in the valuation of public utilities and in adjusting disputes on rates and charges by utilities supplying water and other services. During the World War he was a member of the Pittsburgh valuation board, appraising the street railways in that city. He had also been a member of the valuation committee handling street-railway adjustments in Philadelphia and was consulting expert for the International Railways Company, owner of the street-car system in Buffalo, N. Y., and its suburbs on both sides of the Niagara Valley.

In addition to his consulting work Mr. Fuller found time to write many papers in his field, and three books: "Water Purification at Louisville," published in 1898; "Sewage Disposal," 1912; and "Solving Sewage Problems," 1926, of which his partner, James R. McClintock, was co-author.

At the time of his death Mr. Fuller was chairman of the board of The Engineering Foundation. He was appointed a member of the board by the American Society of Civil Engineers in 1931, served as vice-chairman the following year, and was elected chairman in 1933. He was a past-president and director of the American Society of Civil Engineers and a past-president of the American Water Works Association and the American Public Health Association. He became a member of the A.S.M.E. in 1910 and belonged to many other professional and scientific societies, including the Institution of Civil Engineers, Verein Deutscher Ingenieure, Association Generale des Hygienistes et Techniciens Municipaux, of France, Engineering Institute of Canada, The Franklin Institute, American Chemical Society, American Institute of Consulting Engineers, and the American Society of Bacteriologists. He was chairman of the Committee on Promotion and Attendance for the World Engineering Congress at Tokio, Japan, in 1929.

Among Mr. Fuller's clubs were the Engineers' and Machinery, New York; University, Chicago; Cosmos, Washington; Seaview Golf, Absecon, N. J.; Everglades, Palm Beach; Canoe Brook Country, Summit, N. J.; and the Pleiades Club, New York, of which he was president at the time of his death.

He is survived by his widow, Eleanor Todd Fuller; three sons by previous marriages: Myron E., Kemp G., and Asa W. Fuller; and three step-sons, Kenneth B., Gordon B., and George B. Fuller; and a sister, Mrs. Carl W. DeVoe, of Jerome, Idaho.

THAYER PRESCOTT GATES (1879–1934)

Thayer Prescott Gates, manager of the Finishing Department of the Riverside and Dan River Cotton Mills in Danville, Va., died suddenly of angina pectoris on the evening of January 1, 1934, while in Richmond, Va.

Mr. Gates was born in Lowell, Mass., on May 14, 1879, the son of Prescott Coburn and Ellen (Kittridge) Gates. He prepared for college at Phillips Andover Academy and took a mechanical engineering course at the Massachusetts Institute of Technology. He was vice-president of his class there one year and a member of Sigma Chi fraternity.

Textile manufacturing had been Mr. Gate's work ever since he left the Institute. After some early experience under his father at the mill of Josiah Gates & Sons, in Lowell, he was employed on the construction of mills and layout and erection of machinery at the Lawrence, Mass., mills of the American Woolen Company in 1902–1903. During the next year he secured drafting-room and shop experience with the Jeffrey Manufacturing Company, Columbus, Ohio.

In February, 1904, Mr. Gates became master mechanic for the Wilkes-Barre (Pa.) Lace Manufacturing Company, where he remained for two years. He then took the position of foreman of the J. and W. Lyall Machine Works and master mechanic of the Brighton Mills at Passaic, N. J. In 1907 he became efficiency engineer for the Sayles Bleacheries at Saylesville, R. I., and continued with that mill and the Glenlyon Dye Works, reorganizing them along the lines of scientific management and handling all engineering problems in connection with construction, power supply, maintenance, and operation until 1917. The following year he was connected with Collins and Aikman, of Philadelphia, manufacturers of plushes.

In the fall of 1918 he opened an office in Providence, R. I., as a consulting engineer and textile specialist. He transferred his office to New York in 1930, and continued in practice until 1932, when he took the position he held at the time of his death.

Mr. Gates became a junior member of the A.S.M.E. in 1906 and a member in 1913, and was a Knight Templar in the masonic fraternity. He was a contributor to the textile press and had worked on an emulsifier for use in textile manufacture.

Surviving Mr. Gates are his widow, Cecile (Lovell) Gates, whom he married in 1904, and a daughter, Marjorie (Gates) Alley.

FREDERICK TAYLOR GAUSE (1866–1934)

Frederick Taylor Gause, for nearly forty years connected with the Standard Oil Company of New York, died in San Diego, Calif., on October 17, 1934.

Mr. Gause was born at Kennett Square, Chester County, Pa., on March 16, 1866, son of S. Irwin and Edith (Taylor) Gause. After leaving public school he served an apprenticeship with the Harlin & Hollingsworth Co., and secured drafting and shop experience with that company and at the Trenton Iron Works. He then entered Stevens Institute of Technology, from which he was graduated in 1891.

From the time of his graduation until his retirement in 1930, Mr. Gause was in the employ of the Standard Oil Company. The first few years were spent mainly on experiments to determine the merits of pure petroleum oil for lubricating reciprocating marine engines. In 1894 he was sent to the Far East to report on the possible demand for heavy petroleum products in the principal commercial centers there, particularly at the seven centers in India, China, and Japan where the company had opened offices the previous year to sell kerosene oil. As a result of his investigation the company added to these offices organizations to market heavy petroleum products, and Mr. Gause was assigned to supervise their establishment. Later he made similar trips to South America, South Africa, Australia, New Zealand, etc., establishing marketing branches at many points and otherwise assisting in the conduct of the Export Marketing Department of the company.

Mr. Gause became a member of the A.S.M.E. in 1893, belonged to the Engineers' Club, New York, and was a former member of the American Association for the Advancemnt of Science. He is survived by his widow, Gertrude (Grier) Gause.

FREDERICK AUGUST GEIER (1866–1934)

Frederick August Geier, president of the Cincinnati Milling Machine Company, Cincinnati, Ohio, died suddenly of a heart attack at his home in that city on March 27, 1934.

Mr. Geier was born in Cincinnati on June 23, 1866, the son of Philip and Sophia L. (Otten) Geier. He was graduated from the Woodward High School there in 1884, and for a time worked in a bank in Newton, Kansas. At the age of 21 he became associated with the Cincinnati Screw & Tap Co., which in 1889 became the Cincinnati Milling Machine Company. He served as manager of the company for a time, was made secretary and treasurer, and finally became president.

Within a few years after Mr. Geier's affiliation with the company, increased business necessitated the removal of the plant from Second and Plum Streets, where it had been established in 1884, to new quarters at Spring Grove Avenue and Alfred Street. By this time the company had entered the foreign trade field and exhibited a revolutionary milling machine at the Paris World Fair in 1900.

After 19 years at Spring Grove Avenue and Alfred Street, the company was housed in a plant at Oakley where Mr. Geier was a moving spirit in organizing a factory colony that did much to make the suburb one of the leading machine-tool centers of the world.

Mr. Geier was also president of the Factory Power Company, Factory Colony Company, Cincinnati Grinders Incorporated, Cincinnati Rubber Manufacturing Company, and Cincinnati Morris Plan Bank. He was a director of a number of other companies and institutions, including the Cincinnati Bickford Tool Company, Central Trust Company, Lincoln National Bank, and the Children's Home, all of Cincinnati. He was a leader in promoting vocational education in Cincinnati and aided in the establishment of the cooperative engineering system at the University of Cincinnati when he was a trustee and president of the board of the university. He also established a loan fund for cooperative engineering students. At the time of his death he was president of the Ohio Mechanics Institute.

From 1913 to 1915 Mr. Geier was chairman of the Council of Social Agencies for Cincinnati, from 1915 to 1917 chairman of the Central Budget Committee, and from 1917 until his death, a member of the Executive Committee of the Cincinnati Community Chest.

Other organizations in which Mr. Geier was actively interested included the Cincinnati Bureau of Governmental Research, the National Crime Commission, and the Ohio Valley Improvement Association.

Mr. Geier became an associate of the A.S.M.E. in 1900. He served as chairman of the Cincinnati Section in 1917 and was a manager of the Society from 1917 to 1920. He was interested in the standardize tion and research work of the Society and contributed to the support of the work of the Special Research Committee on Gears.

Mr. Geier was an early advocate of the eight-hour day and helped to organize the National Metal Trades Association. In 1910 and 1911 he was president of the National Machine Tool Builders' Association and in 1933 he helped to organize the Machinery and Allied Products Institute. He was also a member of the American Academy of Political and Social Science, American Institute of Banking, National Municipal League, National Society for Promotion of Industrial Education, Chamber of Commerce of the United States, Cincinnati Chamber of Commerce, and many other such organizations. His clubs included the Engineers, Commercial, Queen City, Camargo, and others in Cincinnati and vicinity.

Surviving Mr. Geier are his widow, Juliet (Esselborn) Geier, formerly of Portsmouth, Ohio, whom he married in 1904, and two sons and two daughters.

HAROLD LOTHROP GOODWIN (1897–1933)

Harold Lothrop Goodwin was born in Roxbury, Mass., on November 14, 1897, son of Homer and Mabel A. (Bates) Goodwin. He was graduated from the Roxbury Latin School in 1916 and received an S.B. degree in mechanical engineering from the Massachusetts Institute of Technology in 1921. Following his graduation he engaged in construction work in Boston with L. P. Soule, soon securing a position with Stone & Webster, Inc. In 1924 he was sent to Philadelphia, Pa., in connection with the construction of a building for the Insurance Company of North America, and in the following year worked on a power-plant building for the Savannah Light & Power Co., and on construction at Davis Islands, Tampa, Fla.

He returned to Boston in 1928 and was associated with L. P. Soule again in constructing an addition to the John Hancock Life Insurance Company building there. In 1930 he took a position in the Generating Office Department of the Edison Electric Illuminating Company of Boston, where he was employed at the time of his death on December 4, 1933.

Mr. Goodwin became a junior member of the A.S.M.E. in 1921. He is survived by his widow, the former Miss Martha L. Dewey, of Great Barrington, Mass., whom he married in 1925.

SAMUEL MARTIN GREEN (1864–1934)

Samuel Martin Green was born at Benton Harbor, Mich., on April 13, 1864, son of Martin Green, a civil engineer well-known for his connection with some of the greatest engineering projects in the United States. His mother was Mary Frances Stuart (Hathaway) Green.

Mr. Green was graduated from the course in mechanical engineering at Worcester Polytechnic Institute in 1885. Following his graduation he was draftsman for about a year with each of the following companies: F. E. Reed Company, Worcester, Mass., machine-tool manufacturers; the John T. Noye Manufacturing Co., Buffalo, N. Y.; the Deane Steam Pump Company, Holyoke, Mass.; and the Merrick Thread Company, Holyoke. He then became master mechanic of the last-named company, which permitted him to do outside consulting work. About 1898 the American Thread Company was formed and the Merrick Thread Company merged with it. He served as consulting engineer to the company until he organized his own business in the fall of 1908 at Springfield, Mass., under the firm name of Samuel M. Green Company.

Among the numerous companies which Mr. Green served as consulting engineer prior to the establishment of his own company were the Union Metallic Cartridge Company, Bridgeport, Conn., and the United States Envelope Company. For the latter company he designed and supervised the construction of a three-story factory building at Waukegan, Ill., and a five-story structure at Springfield in connection with which he designed and supervised the installation of a steam boiler and power plant and electrified the entire plant.

In connection with Mr. Green's work for the Union Metallic Cartridge Company, the following is taken from a memoir prepared by Harry B. Hopson, of Springfield, for the American Society of Civil Engineers, of which Mr. Green was also a member:

"In 1906, Mr. Green designed a steam boiler and turbine plant for the Union Metallic Cartridge Company. When this work was completed, the company requested him to make an examination of its primer loading plants after a disastrous explosion. He revised the method of loading primers entirely; built a new primer-loading plant; and devised an entirely new method of handling and mixing fulminate, which, to that time, had been an exceedingly dangerous operation. Since the revision of the method by Mr. Green the company has never had an explosion in its plant.

"About that time, the company had a disastrous explosion in its powder-storage farm where many tons of powder had been stored. Mr. Green was given the problem of designing a powder-storage farm that would be safe. He designed roads, bunkers, and small storage houses for powder, and supervised the installation of this equipment.

The Union Metallic Cartridge Company was the owner of the Remington Arms Company, at Ilion, N. Y. This latter company was not making a profit, and Mr. Green was given the commission of investigating the entire process of manufacturing arms. He spent two or three years at the Remington plant, designing new equipment and re-arranging machinery, until, finally, it was operating on a profitable basis."

Concerning Mr. Green's work after 1908, Mr. Hopson writes: "Among his more important clients was the Farr Alpaca Company, of Holyoke, which spent under his direct supervision and charge from $12,000,000 to $15,000,000.

"In 1911, he was given a commission by Max Henkels, who had come to the United States from Germany to engage in the lace-manufacturing business, at Bridgeport. Mr. Green designed and supervised the construction of a weave shed; a two-story and basement brick mill building; and in 1915, a three-story building of reinforced concrete. The design of this latter building was rather extraordinary because the posts had to be located so that Nottingham-lace looms could be installed in the building. The floors of the building were of the mushroom type of construction.

"In 1912 and 1913, Mr. Green designed and supervised the construction of a paper mill for the Strathmore Paper Company, at Woronoco, Mass., together with its power plant, filter plant, etc. For this same company, in 1923, he designed a reinforced-concrete bridge across the Westfield River, a span of approximately 400 ft, supported by three high arches.

"In 1915, he designed and supervised the construction of the plant of the Warner-Klipstein Chemical Company, at South Charleston, W. Va. This was a caustic-chlorine plant consisting of many buildings, including a power plant of about 2000-kw capacity, a plant for the production of carbon tetrachloride, carbon bi-sulphide, chlorine, and caustic soda.

"In 1918, Mr. Green was awarded the contract by the United States Government to design and supervise the construction of the large chlorine plant at the Edgewood (Md.) Arsenal. This chlorine plant was the largest of this character that had ever been built to that time. Its capacity was 100 tons of chlorine gas and caustic soda per 24 hr. Mr. Green's work consisted in designing all the buildings which were of brick and wood construction. This plant consisted of a building for chlorine cells; a salt-treating plant; a boiler plant, of approximately 2000 boiler-hp; an evaporator plant for salt caustic evaporation, containing eight double-effect evaporators; and a caustic fusion plant. He designed all the machinery and supervised its installation, and was under contract with the Government to supervise the installation of all the equipment and the starting of its operation. This plant was successfully constructed, and operation was started before the end of the World War, or in approximately six months.

"In 1926, Mr. Green designed a new boiler plant for the Fitchburg Paper Company, of Fitchburg, Mass. This work consisted of a complete revision of an old boiler-house and the installation of one 1000-hp, 250-lb pressure boiler, with complete pulverized-coal equipment, superheaters, and piping system. In connection with this work he also designed a high-pressure steam line, with all the supports, about 1200 ft long, running from this boiler plant to one of the other mills over a bridge, about 100 ft long, also designed by Mr. Green. This work was entirely under his supervision; he let all the contracts and supervised the completion of the work.

"In 1927, he designed a new boiler installation for Joseph Bancroft and Sons Company, at Wilmington, Del. This work consisted in the removal of two old boilers and the installation of one 1000-hp, 250-lb pressure boiler, together with all its auxiliary equipment, stokers, etc., and a revision of the boiler-house for this installation.

"During 1927 and 1928, Mr. Green also engineered and supervised the construction of the Integrity Trust Building in Philadelphia, Pa."

Mr. Green had been a member of the A.S.M.E. since 1890, and in addition to his membership in the A.S.C.E., belonged to the Engineering Institute of Canada, Engineering Society of Western Massachusetts, and the American Geographical Society. He was a member of the Chamber of Commerce and Taxpayers' Association of Springfield, and president of the latter at the time of his death. He also belonged to the Constitutional Liberty League of Massachusetts, National Economy League, and Navy League of the United States. He was a member of all the branches of the York Rite and of the Scottish Rite of the masonic fraternity, being a Knight Templar, a Thirty-second Degree Mason, and a member of the Shrine.

Mr. Green's death occurred at Springfield on March 22, 1934. He is survived by his widow, formerly Miss Ida McKown, of Holyoke, and by their two married daughters, Mrs. Mildred (Green) Page, of Longmeadow, Mass., and Mrs. Lydia (Green) Meadows, of New York, N. Y.

JOSEPH RUDD GREENWOOD (1883–1934)

Joseph Rudd Greenwood, who retired in 1930 as president and treasurer of the New York Vitreous Enamel Products Corporation, Flushing, L. I., N. Y., died in New York on March 2, 1934.

Mr. Greenwood was born in New York on May 27, 1883, the son of Isaac J. and Mary A. (Rudd) Greenwood. He attended the Collegiate and Halsey schools in New York and the Hill School at Pottstown, Pa., and was graduated as a civil engineer from Princeton University in 1905. After serving as an instructor in civil engineering at Princeton in 1905–1906, he took a position as assistant engineer with The Ballwood Company in New York, manufacturers of single- and four-valve engines, high- and low-pressure and mixed-flow turbines, and high-pressure pipe joints and bends. He was made superintendent of the pipe shop in 1908, superintendent of construction in the field in 1910, general superintendent in 1911, and general manager in 1914.

In 1916 Mr. Greenwood left The Ballwood Company to be mechanical engineer in the office of Charles H. Higgins, but the following year entered the World War as a driver and section leader in the American Ambulance Field Service in France. He rose to the rank of captain in the service.

After the close of the War he returned to the employment of Mr. Higgins, with whom he remained for several years before becoming connected with the New York Vitreous Enamel Products Corporation.

Mr. Greenwood became a member of the A.S.M.E. in 1920, and belonged to the Sons of the Revolution, and the University and Princeton clubs. He is survived by his widow, Ruth M. (Dayton) Greenwood, whom he married in 1919.

ELWOOD GRISSINGER (1869–1934)

Elwood Grissinger, of Buffalo, N. Y., whose death occurred on October 8, 1934, was born at Mechanicsburg, Pa., on March 3, 1869. After completing his high school course there he studied telegraphy in the ticket office of the Cumberland Valley Railroad at Mechanicsburg. At the age of 17 he secured an appointment as a telegraph operator on the Norfolk & Western R.R. Subsequently he studied stenography at Chaffee Institute, Oswego, N. Y., and worked in the general passenger office of the New York, Ontario & Western R.R. and for the Pennsylvania Railroad, at Harrisburg, Pa.

In 1890 Mr. Grissinger entered Lehigh University, from which he was graduated four years later with a degree in electrical engineering. His scholastic standing was such that he was appointed orator for Founder's Day and commencement. He was elected to membership in Tau Beta Pi, honorary engineering fraternity, as well as in Delta Upsilon. In 1916 the university conferred an honorary M.S. degree upon him.

After his graduation from Lehigh University Mr. Grissinger worked for two years in the shops of the Westinghouse Electric & Manufacturing Co., and for three years was district engineer and salesman for the company at Syracuse and Buffalo. From 1899 to 1906 he was commercial engineer for the Buffalo General Electric Company and the Cataract Power & Conduit Co. The following year he was general agent for the Niagara Falls Power Company and the Canadian Niagara Power Company.

Since 1906 Mr. Grissinger had engaged in consulting engineering in the mechanical and electrical fields, and in research work. He patented a complete system of long-distance telephony, including telephonic speech amplifiers and the only known operable method of associating them with the speech transmission circuits. This system has been adopted for use on transcontinental lines and all classes of long-distance circuits, the United States patents having been purchased by the American Telephone & Telegraph Co., and was used in Europe by the United States Army and its Allies during the World War. He also patented a method of sound amplification as exemplified by phonograph horns.

Mr. Grissinger became a member of the A.S.M.E. in 1912. He was a Fellow of the American Institute of Electrical Engineers and a member of The Franklin Institute. He had contributed special articles on electric railroading to the journals of the Brotherhood of Locomotive Engineers and Firemen.

Mr. Grissinger married Lucy Martin Ash, of Oil City, Pa., in 1897, and is survived by her and a son, Elwood.

CONRAD VELDER HAHN (1890–1933)

Conrad Velder Hahn, associate professor of mechanical engineering at Drexel Institute, Philadelphia, Pa., and head of the mechanical engineering department of the Drexel Evening Diploma School, died on December 3, 1933, at the Temple University Hospital.

Dr. Hahn was born in Philadelphia on May 9, 1890, the son of John G. and Elizabeth (Velder) Hahn. He attended the Central Manual Training High School in Philadelphia and was graduated from the University of Pennsylvania with a B.S. degree in engineering in 1908. An M.E. degree was conferred upon him by the same university four years later and San Jose University gave him the degrees of Doctor of Engineering and Doctor of Philosophy.

Dr. Hahn was a member of the faculty of the civil engineering department of Temple University Evening School from 1910 to 1912 and of the mechanical engineering department of the University of Pennsylvania during the next two years. He had been connected with Drexel Institute since 1919, when he became director of the mechanical engineering department of the evening school.

He had also carried on a consulting practice since 1910 as managing director of the Hahn Company, Philadelphia. He was also a director and associate engineer of the International Development Company, Washington, D. C., consulting mechanical engineer for the Turbo Motor Company, East Radford, Va., and a member of N. T. Whitaker & Co., Washington.

During the World War he served in the Ordnance Department, as first lieutenant and captain, and received special mention for technical service at the Army Ordnance Base Depot, France, and the Aberdeen Proving Ground, Scotland. He held the rank of major in the Ordnance Reserve Corps.

Dr. Hahn became a junior member of the A.S.M.E. in 1915. He also belonged to The Franklin Institute, National Philosophical Society, masonic fraternity, and a number of fraternal orders and clubs, including the honorary engineering fraternities, Sigma Xi, Tau Beta Pi, and Pi Tau Sigma.

He is survived by his widow, Lillian (Roth) Hahn, and by his mother, a brother, and a sister.

THEODORE COMMODORE HAILES, JR. (1883–1934)

Theodore Commodore Hailes, Jr., field engineer for the New York State Temporary Emergency Relief Administration, Binghamton, Division Office, died in that city on October 29, 1934.

Mr. Hailes was born in Albany, N. Y., on September 15, 1883, the son of Theodore Commodore and Elizabeth (Brooks) Hailes. After his graduation from the Albany High School he entered Rensselaer Polytechnic Institute, from which he received a C.E. degree in 1907. He took part in practically all extra-curricula activities while at the Institute and was one of the founders of the Rensselaer Technical Society, which later became Pi Kappa Phi fraternity.

From the time of his graduation until the United States entered the World War, Mr. Hailes engaged in the general practice of engineering and contracting, with an office in Albany. His work included examinations and reports on a number of hydroelectric-development projects; the design and installation of signal equipment on the Erie Canal; construction of several telephone lines; destruction by dynamite of ice gorges in the Hudson River several years; supervision of construction of state highways, and quarrying and crushing of stone for them; and investigation and reports as expert witness in many cases involving engineering.

In 1917 he accepted a position as superintendent of the Neemes Brothers Iron Foundry, at Troy, N. Y., which was engaged in war work. He continued with that company for several years after the close of the war, then became manager for the Tool & Device Corp., of Troy. More recently he had been secretary and treasurer of the Engineering Service Corporation, Albany.

Mr. Hailes became a member of the A.S.M.E. in 1922 and was a 32d degree Mason. He is survived by his widow, Charlotte S. (Welch) Hailes, whom he married in 1909, two sons, Theodore C. Hailes, 3d, and Gardner M. Hailes, and three sisters.

FRANCIS EDWARD HALLORAN (1882–1932)

Francis Edward Halloran was born on June 5, 1882, in Brooklyn, N. Y., the son of Patrick and Anne (Burke) Halloran. He was graduated in 1900 from the Boys High School in Brooklyn and later continued his studies through the International Correspondence Schools.

The greater part of Mr. Halloran's engineering work was for the E. W. Bliss Co., Brooklyn. He was associated with this company from early in 1902 until after the close of the World War with the exception of about a year during which he worked for C. J. Regan, Maspeth, L. I., designing dies for sheet-metal ware and attachments for spinning lathes. His first assignments for the Bliss company were in the general drafting room. He was then transferred, upon his request, to the drafting room of the die department. After the year at Maspeth he began his most important work for the company, in connection with torpedoes. In the capacity of mechanical draftsman in the torpedo drafting room at the main works of the company he worked from 1906 to 1910 laying out the mechanisms in the Bliss-Leavitt torpedo; helping to develop the outside superheater system; experimenting with gyroscopic steering, hydrostatic depth regulating, and reducing-valve gears; redesigning dynamometers; and developing new tools and fixtures for the manufacture of torpedo parts. He was made chief draftsman in 1910 and during the next three years was engaged in the design and construction of an "angle-fire" gyroscope, depth mechanism, steering engines, launching tubes, and high-pressure air accumulators for the company's proving steamer *Emblane*.

In 1913 the torpedo department was relocated at the 53d Street works of the company and for about ten years longer Mr. Halloran devoted himself to supervising all the experimental and research work incident to further increasing the speeds, range, and reliability of torpedoes.

After leaving the Bliss company Mr. Halloran was mechanical designer first at the Edison Lamp Works of the General Electric Company at Harrison, N. J., and then with the Trans-Lux Daylight Picture Screen Corporation, Brooklyn. He helped to perfect the machinery for manufacturing frosted bulbs, in the former position, and in the latter worked on the Trans-Lux stock quotation projection machine and motion picture projection. Poor health forced his retirement from the Trans-Lux company, and his death from heart disease occurred on March 20, 1932, at the home of his sister, Mrs. M. Gertrude Quinn, Corona, L. I., N. Y.

Mr. Halloran became a member of the A.S.M.E. in 1920.

JOHN HANKIN (1865–1933)

John Hankin, for nearly forty years president of John Hankin & Brother, consulting engineers, New York, N. Y., died at his home in Passaic, N. J., on November 30, 1933.

Mr. Hankin was born on September 21, 1865, in Farnworth, Lancashire, England, the son of William and Susannah Hankin. He came to the United States with his parents in 1874 and secured his early education in the public schools in Rome and Watertown, N. Y. Subsequently he attended the Brooklyn Evening High School for one year and took mechanical and scientific courses at Cooper Institute.

Mr. Hankin's first employment was in Watertown, N. Y., in the shops of the Davis Sewing Machine Company, in 1880. He then served a four-year apprenticeship at the Steam Engineering Works there, after which he was employed as a journeyman machinist at the works and by the Eames Vacuum Brake Company, Watertown, the James Brady Manufacturing Company, Brooklyn, and Arnoux & Hochausen Electric Co., New York. For two years, beginning in March, 1888, he was draftsman for the Buffalo Forge Company, and for the next three years engaged in sales engineering for the Akron Heating & Ventilating Co., Akron, Ohio, and Cook & Co. and Jas. Beggs & Co., New York.

From August, 1893, to a year from that time Mr. Hankin was a partner in the Armstrong Engineering Company, New York. He then went into the general engineering business for himself. The firm of John Hankin & Brother was established in 1900.

Mr. Hankin became a member of the A.S.M.E. in 1913 and belonged to the National Association of Heating and Piping Contractors, American Society of Professional Engineers, American Institute of the City of New York, and Building Trades Employers' Association of New York. He was also a member of the masonic fraternity, the Blizzard Men of 1888 Club, and the Transportation Club, New York.

Surviving Mr. Hankin are his daughter, Mrs. E. Hans Jacob, of New York; a brother, Dr. Walter Hankin, of Bisbee, Ariz.; a stepson, Dr. John Townsend, of Passaic, a son of his deceased wife, the former Mrs. Ida Dieter Townsend, of Buffalo; and two grandsons, Robert Earl and John Edward Jacob.

JAMES HARTNESS (1861–1934)

The American Society of Mechanical Engineers lost one of its most honored members, and the profession one of its most distinguished engineers, in the death of Ex-Governor James Hartness, of Springfield, Vermont, after a long illness, on February 2, 1934. His loss will be deeply felt by all who have ever known him.

James Hartness was born in Schenectady, N. Y., on September 3, 1861. His father, John W. Hartness, a man of dry, salty humor and keen understanding of human nature, moved to Cleveland in 1863, where he was a foreman in various machine shops. James Hartness was educated in the Cleveland public schools and began learning his trade at sixteen, at a wage of 45 cents a day, first with Younglove, Massey, and Company, of Cleveland, where his father was superintendent, and then in the machine shop of the Union Steel Works. In the latter shop he first came into contact with close, accurate work

under Jason A. Bidwell, one of the fine New England mechanics of the older generation. Later, Mr. Hartness went to the Lake Erie Iron Works as a toolmaker. In 1882 he secured a position, through correspondence, as foreman of the Thomson, Stacker Bolt Company in Winsted, Conn. When he reached there the superintendent exploded at the youth of the man whom he had hired. Hartness waited until the storm was over and then offered to release the firm from its contract but to stay until his successor was installed. He remained three years. In 1885 he went to the Union Hardware Company, of Torrington, Conn., manufacturers of gun implements, first as toolmaker, then foreman, and later as inventor. During the year 1888 he worked for a few months each at the Pratt & Whitney Company in Hartford, at Scottdale, Pa., and with the Eaton, Cole, and Burnham Company in Bridgeport.

Connection With Jones & Lamson Machine Company Established in 1889

He went to the Jones & Lamson Machine Company in March, 1889, the year in which the company moved from Windsor to its present location at Springfield, Vermont. He was at first superintendent until 1893, then manager until 1900, and president from then until his retirement last year. Mr. Hartness' going to the Jones & Lamson Machine Company added another to the list of great mechanics who made the history of this company, or rather the succession of companies of which it is a part. It began about 1838 at Windsor, Vermont, as the Robbins and Lawrence Company, and through the middle of the century, under Richard S. Lawrence, Frederick W. Howe, and Henry D. Stone, this small shop made engineering history. The ever-widening influence of their work is felt to this day. Here was developed and first manufactured the hand-operated turret lathe, the beginnings of the milling machine later known as the Lincoln miller, drop hammers, and many tools adapted to arms manufacture and other types of accurate work. Through various changes the firm became Lamson, Goodnow, and Yale, E. G. Lamson and Company, the Windsor Manufacturing Company, and in 1879 the Jones & Lamson Machine Company. Mr. Hartness brought with him the fundamental ideas of the Hartness flat turret lathe which he had had in his mind for a number of years. He undertook, as superintendent, to develop and manufacture this machine, and in so doing opened up a new period in the history of the company. Like other shops of the time, the successive companies at Windsor had scattered their energies over a wide line. An old "poster" of the Windsor Manufacturing Company shows them offering engine lathes, turret lathes, planers, saw-mill and quarrying machinery, engines and boilers, trip-hammers, sewing machines, rifles, "etc., etc." Things had improved somewhat in this respect, but with the removal of the company to Springfield and the coming of Mr. Hartness there was a radical change. He concentrated its activities on a single size of the flat turret lathe, and the tools for it, dispensed with agents and sold direct, and instituted a new type of sales literature. Under his leadership the company rapidly acquired a world-wide recognition. This is all the more remarkable for its being located in a small town in a Vermont valley six miles off a minor railway, a location, by the way, which he always defended and said he would in no wise change.

Services to A.S.M.E. and the Public

Mr. Hartness joined The American Society of Mechanical Engineers in 1891. He was a manager from 1909 to 1912, and vice-president in 1912 and 1913. On the German trip of the Society in 1913, President Goss was unable to go and Mr. Hartness, as senior vice-president, acted as head of the party on all the formal occasions. He did this so acceptably to all that he was elected president of the Society in the following year, 1914. He was president of the American Engineering Council from 1924 to 1926, and as a past-president, served as a member of its Assembly for the six years following his presidential term. He therefore held the highest elective positions in the gift of his profession.

In 1912 he was appointed a member of the A.S.M.E. Special Committee on Tolerances in Screw-Thread Fits, and he served for many years on this and other committees on screw-thread standardization.

His public services were as distinguished as those for the engineering profession. He served his state as chairman of the State Board of Education for six years (1914–1920), was chairman of the Committee of Public Safety during the War, and also Federal Food Administrator, and, in 1921 to 1923, served as the sixty-first governor of Vermont. In national affairs he was a member of the commission which represented the United States at the Inter-Allied Air Craft Standardization Conferences in London and Paris; and was vice-chairman of the Congressional Screw-Thread Standardization Committee from 1918 to 1920.

He was a member of the Society of Automotive Engineers, the Aero Club of America, the American Astronomical Society, a Fellow of the American Association for the Advancement of Science, the Royal Astronomical Society, the Royal Society for Encouragement of Arts, the Royal Societies Club of London, and the Institution of Mechanical Engineers in London. In 1921 he was awarded the John Scott Medal by the Board of Directors of the City Trusts of Philadelphia, and the Edward Longstreth Medal by The Franklin Institute of Philadelphia.

He is the author of a number of papers before the A.S.M.E., including his presidential address on "The Human Element, The Key to Economic Problems." He also wrote two books, "The Human Factor in Works Management" which was widely distributed, and translated abroad, and "Machine Building for Profit."

He was given the honorary degree of M.E. by the University of Vermont in 1910, honorary M.A. by Yale University in 1914, and LL.D. by the University of Vermont in 1921.

In 1885 he married Lena Sanford Pond, of Winsted, Conn., who died in March, 1933. He is survived by two daughters, Anna Jackson, who married Dr. William H. Beardsley, and Helen Edith, who married Mr. Ralph E. Flanders.

Mr. Hartness a Prolific Inventor

Mr. Hartness was a prolific inventor and he held patents in this country and abroad for more than 100 inventions. Among these was the flat turret lathe which he began developing from the time of his first coming to Springfield. In the old type of high turret lathes, developed originally by the Robbins and Lawrence Company, the turret revolved about a large vertical pin and all subsequent development had followed that design. Mr. Hartness felt that a stiffer and more accurate construction was possible and investigated the disk turret, the barrel turret, and the flat turret. He settled finally upon the flat turret which appealed to him as offering all the advantages of the old type with possibilities of greater stiffness and precision. It was low, could be clamped at points far apart, and permitted the location of the locking pin directly under the cutting tool. Following out this general design, he developed a lathe which has found wide use here and abroad for the large output of accurate work. Among the improvements were the cross-sliding head in 1903, the double-spindle type in 1910, and the Hartness automatic lathe.

The Lo-Swing and Fay automatic lathes are two more machines invented by him, or greatly influenced by his work. In connection with lathes, he developed many special tools to be used in connection with them, and among these were the well-known Hartness threading dies, which became an important element in the company's business. Associated with these was the Hartness thread comparator, a refined method for the gaging of screw threads by optical means, which not only tells whether the threads are off standard, but precisely wherein they depart from it. He also invented the Hartometer, another type of gage for screw-thread inspection.

His Interest in Astronomy

Mr. Hartness' avocations were as interesting as his vocation. He spent four summers on a yacht following down the coast as far as Chesapeake Bay. In 1916 he learned to fly and won an amateur pilot's license. His interest in aviation continued throughout his whole life and he was largely influential in establishing a landing field at Springfield as a memorial to the soldiers and sailors of the World War. It was at this field that Lindbergh landed for his stop in Vermont during the tour of the country after his Paris flight.

For years astronomy claimed his attention. After suffering from a cold acquired in using the old-fashioned type of telescope, he devised the turret equatorial telescope, which was an adaptation of the fundamental idea of the flat turret lathe, wherein a large ring was substituted for one of the axes in the usual equatorial type. This permitted the observer to work inside, under warm and comfortable conditions at all times. One of the happy incidents of the A.S.M.E. trip in 1913 was to run into a model of this type of telescope in the astronomical section of the Deutsches Museum at Munich. This invention was the subject of a valuable paper before the Society at its Annual Meeting in 1911, and gained for him membership in both English and American astronomical societies. It was mounted over an underground room, one of a long series, running under the lawn in front of his home, connected by a heated and air-conditioned tunnel, and containing workrooms, library, and lounging room. Here he did much of his later work, and it formed a unique feature of the home whose charming hospitality will long be remembered.

His interest in astronomy aroused many others, so that there is probably no community of size similar to Springfield where there is such widespread interest in astronomy and in the building of amateur telescopes.

His Influence on Other Men

An outstanding characteristic of Mr. Hartness' life was his influence on and encouragement of young men. A number of impor-

tant machines have been developed by men connected with his company and are now being manufactured by other firms. He discovered and helped in various ways many men now well known in the engineering profession, among whom are George O. Gridley, inventor of the Gridley automatic, and vice-president of the New Britain-Gridley Machine Company; Edwin R. Fellows, inventor of the gear shaper, and president of the Fellows Gear Shaper Company; the late William L. Bryant, who was president of the Bryant Chucking Grinder Company; Frederick P. Lovejoy, president of the Lovejoy Tool Company; and Ralph E. Flanders, who succeeded Mr. Hartness as president of the Jones & Lamson Machine Company. Mr. Hartness' concentration on his own machines was so great that he would not be diverted in his own work by the inventions and ideas of these younger men, but as they brought their ideas to him he freely gave them help and encouragement and was a factor in their success.

Though Springfield is a small country town, there has grown up there a large group of important tool-building plants, largely through his influence.

Governor Hartness was one of the foremost mechanical engineers of this country and generation, and a splendid example of the engineer in public life. The Society holds him in affection and honor as an engineer and as a citizen.—J. W. Roe.[1]

WILLIAM JOSEPH HENNESSY (1873–1934)

William Joseph Hennessy, a veteran of the Spanish-American War, died in Washington, D. C., on November 26, 1934, and was buried in Arlington Cemetery with military honors. When war was declared with Spain Mr. Hennessy was employed by the American Line Steamship Company on the *St. Paul*, which was taken over by the Navy Department and placed under the command of Captain Charles D. Sigsbee, who had commanded the battleship *Maine*. Mr. Hennessy enlisted as a chief machinist and served on the *St. Paul* until the close of the war.

He was born at Jersey City, N. J., on September 27, 1873, son of Michael J. and Ellen T. (Clancy) Hennessy. His father was chief engineer in the United States Navy. After he had completed his high-school course in Jersey City he became an apprentice machinist with the William Cramp & Sons Ship & Engine Building Co. and during his apprenticeship also took evening courses in mechanical engineering at The Franklin Institute, and correspondence courses in marine engineering through the International Correspondence School.

He entered the employ of the American Line in 1895, and with the exception of his six months in the Spanish-American War, was connected with the company for five years, serving from junior to second engineer on vessels between the United States and England. In 1901 and 1902 he was a machinist in the Construction and Repair Department at the United States Navy Yard, New York, engaged for the most part in experimental work. During the next two years he was first assistant engineer for the Mallory Steamship Company, on boats running between New York and Georgia, Florida, and Texas cities.

He returned to Government service in November, 1905, as chief engineer in the United States Coast and Geodetic Survey. During about eight of the twelve years he spent in this service he was connected with important surveying expeditions in the Philippines, Alaska, and the West Indies, as well as work on the Atlantic Coast. Four years were spent in shore duty in the office of the Survey at Washington, and in different parts of the United States. One of his assignments was the inspection of the steam yacht *Isis* in 1915, which was taken over by the Navy upon his recommendation. He designed, wrote the specifications,and supervised the construction of the boilers, engines, and auxiliaries of the steamer *Surveyor*, built in 1917 for the Survey at the works of the Manitowoc Ship Building Company in Wisconsin, and served as chief engineer on its maiden trip to Washington and Norfolk.

During the World War Mr. Hennessy was inspector of engines and boilers for the Emergency Fleet Corporation of the United States Shipping Board, in connection with work being done in the Central States area.

Subsequently he was superintendent, for a time, of marine engine construction for the Dodge Manufacturing Company, in Indiana, and more recently had served as chief engineer in charge of the plants at Woodstock College, Gonzaga College, and St. Aloysius Church, Washington. He was in poor health the last few years of his life.

Mr. Hennessy received medals from the United States Navy and Shipping Board, and from the State of New Jersey for his war service.

Mr. Hennessy was not married. He is survived by a brother, the Reverend Charles J. Hennessy, of Woodstock, Md., and by two sisters, Miss Ella Hennessy and Mrs. Lydia Bahlman, both of Washington. He became an associate-member of the A.S.M.E. in 1918.

WILLIAM THOMAS HENRY (1845–1933)

William Thomas Henry, of Fall River, Mass., died in that city on July 22, 1933. He was born there on February 27, 1845, son of James and Martha (Whitaker) Henry, and received his early education in local schools. He took courses in civil and mechanical engineering with the class of 1870 at the Massachusetts Institute of Technology. He then entered the employ of Josiah Brown, civil engineer, Fall River, and upon his death five years later took over the business, which he continued until his retirement from business. He was connected with the design and construction and engineering problems of many of the textile mills in that part of the country and the inventor of a belt-driving mechanism patented in 1910.

Mr. Henry became a member of the A.S.M.E. in 1887 and belonged to the American Historical Society in Fall River and the Methodist Episcopal church there. His first wife, S. Louise Hadcock, whom he married in 1884, died nine years later. He is survived by his widow, Carrie S. Henry, whom he married in 1898.

FREDERICK WILLIAM HESPE (1889–1934)

Frederick William Hespe, whose death occurred on April 16, 1934, was born in Jersey City, N. J., on July 27, 1889. He attended the Stevens School and Stevens Institute of Technology, Hoboken, N. J. His first position was with the Public Service Corporation of New Jersey, Newark, where he worked on machine design and in the claims department from 1911 to 1918. The next year was spent as chief inspector of metal parts for the Standard Aircraft Corporation, Elizabeth, N. J. In 1919–1920 he was machine designer for the American Sawmill Machinery Company, Hackettstown, N. J., and from 1920 to 1923 held a similar position at the Picatinny Arsenal, Dover, N. J. He became co-partner of the Wharton (N. J.) Foundry in 1923 and owner in 1926, and engaged in the manufacture of gray iron castings until 1928. He then went to the Fokker Aircraft Corporation of America, Passaic, N. J., where he worked first as machine designer and subsequently as supervisor of the shop of the research department. More recently he had been supervisor of research at the General Aviation Manufacturing Corporation, Passiac.

Mr. Hespe became a member of the A.S.M.E. in 1931. He is survived by his widow, Rita Hespe.

FRANK JOSEPH HESS (1896–1934)

Frank Joseph Hess, instructor at Brooklyn (N. Y.) Technical High School from 1926 until the time of his death on December 26, 1934, was born in Buffalo, N. Y., on August 17, 1896. His parents were Frank William and Mary H. (Schroeder) Hess. He secured his early education in the schools of Clifton, N. J., then attended the Stevens Institute of Technology for two years. He secured a B.S. degree from New York University in 1920 and his M.E. degree two years later. Columbia University conferred a posthumous M.A. degree upon him in February, 1935.

Mr. Hess was an assistant superintendent with the Tidewater Oil Company in Bayonne, N. J., from 1920 to 1924. He was an engineer at Columbia University between then and the time he joined the faculty of the Brooklyn Technical High School.

Mr. Hess became a junior member of the A.S.M.E. in 1921 and also belonged to the masonic fraternity. He was an ensign in the United States Naval Reserve in 1918 and a sergeant in field artillery in the United States National Guards, in New York, from 1926 to 1930. He is survived by his widow, Edna M. (Flewelling) Hess, whom he married in 1924.

EDWARD MARRIOTT HEWLETT (1866–1934)

Edward Marriott Hewlett, an electrical engineer and inventor for many years associated with the General Electric Company and known for his connection with important engineering projects, died at his home in Schenectady, N. Y., on May 24, 1934, of a cerebral hemorrhage.

Mr. Hewlett was born at Cold Spring Harbor, L. I., N. Y., on September 14, 1866, a son of Edward T. and Eliza (Marriott) Hewlett. He was educated in private schools and by private study in civil and mechanical engineering and physics. From 1884 to 1890 he was general manager and civil engineer of the Cascade Town Improvement Company, Cascade, Colo. In 1888 he surveyed and laid out the first road to the top of Pike's Peak.

Mr. Hewlett took up electrical engineering in April, 1890, when he

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[1] Professor of Industrial Engineering, New York University, New York, N. Y., and author of "English and American Tool Builders," McGraw-Hill Book Co., 1926. Mem. A.S.M.E.

went to work for the Thomson-Houston Electric Company, at Lynn, Mass., where he assembled, tested, and repaired electrical machinery. After about a year there he was sent to South River, N. J., where he worked until March, 1892, on the design and construction of embroidery machinery. He was then transferred to the New York office of the Thomson-Houston company and assisted in the installation of a focusing lamp and strings of incandescent lamps on the Statute of Liberty for the Columbian Exposition. After this work was completed he returned to Lynn, where he assisted in designing electrical switchboards for railways.

In 1894 Mr. Hewlett went to Schenectady as assistant design engineer for the General Electric Company, which had taken over the Lynn plant. Two years later he was advanced to the position of design engineer in the railway department, and when the switchgear department was organized in 1898 he was appointed its head. He was active in the work of this department until 1927, when he became consulting engineer for it. He retired from service in 1931.

Mr. Hewlett designed switchboards for the control of the locks of the Panama Canal and for the Metropolitan Street Railway Company, New York Edison Company, Manhattan Elevated Railway, Interborough Rapid Transit Company, and Grand Central Terminal, all in New York. He also handled the electric lighting of Tiffany's store in New York and designed special electrical effects for outdoor displays.

Between 1905 and 1907 Mr. Hewlett, in association with H. B. Buck, then engineer of the Niagara Falls Power Company, evolved the Hewlett suspension insulator, which made possible the transmission of voltages between 100,000 and 220,000 and higher, as against about 60,000 volts which could be sent with the earlier pin-type insulator. In 1906 he received a silver medal at the St. Louis Exposition for the development of the oil switch, from which sprang the entire modern line of oil circuit breakers, operating at high interrupting frequencies. He also originated the electrical relay, an indispensable factor in electrical control systems. He received the Coffin Award of the General Electric Company for special marine apparatus. He proposed and designed the modern fire-control system of the United States Navy, and during the World War was chief engineer of the Navy Experimental Station at New London, Conn. Altogether, Mr. Hewlett held about 160 patents for his inventions.

Mr. Hewlett became a junior member of the A.S.M.E. in 1897 and a member three years later. He was a Fellow of the American Institute of Electrical Engineers, and belonged to the masonic fraternity as well as to a number of clubs in New York and Schenectady, including the Engineers' in New York and the Mohawk and Electrical in Schenectady.

Surviving Mr. Hewlett are his widow, Mrs. Susan (Mauger) Hewlett, who resides with her son, John Mauger Hewlett, of Teaneck, N. J.; and two other sons, Edward P. Hewlett, of Schenectady, and Paul Marriott Hewlett, of Albany, N. Y.

LOUIS JOSEPH HIRT (1854–1933)

Louis Joseph Hirt, retired consulting engineer, New York, N. Y., died in that city on September 20, 1933. He was born in Paris, France, on September 26, 1854, and was educated in that country. He served as an apprentice in the shop of his father, Joseph Hirt, for two years and secured further experience in the marine shop of Esher Wyss & Co., Zurich, Switzerland, and with the Northern Railroad of France. He went to London in 1876 and was employed successively by Leger & Co., manufacturers of mathematical and optical instruments, and by J. Moore as general workman on watchmaker's tools. In 1878 and 1879 he was foreman, first at the Keystone Valve Company in Wolverhampton, England, and then at the steam-pump works of J. Evans & Son there.

Mr. Hirt came to the United States in July, 1880, and until the fall of 1882 was employed as foreman at the machine shop of Hanan & Lewes, New York. From then until 1885 he was superintendent of the machine shop of Merrill Bros., Brooklyn. He went to Rochester, N. Y., in 1885 and for two years was master mechanic of the Standard Nail Company there. During the next year he was chief draftsman at the Walker Manufacturing Company, Cleveland, Ohio, and from 1888 to 1890 traveling engineer for the Hill Clutch Works of that city.

From 1890 to 1896 Mr. Hirt was connected with the West End Street Railway, Boston, Mass., as master mechanic and chief engineer. He then returned to New York, taking the position of mechanical engineer for the New York Cable Railroad (later known as the Metropolitan). After two years with that company he became chief engineer for the New England Gas & Coke Co., Everett, Mass.

In 1900 Mr. Hirt took a position as mechanical engineer for the Pearson Engineering Corporation, New York, later becoming its vice-president and continuing in that capacity until 1920. In 1906 he took up consulting work for the Toronto Power Company, Toronto, Canada, and the Winnipeg Electric & Manitoba Power Companies, Winnipeg, Canada, which he served until 1926. In 1910 he also became consulting engineer for the Brazilian Traction, Light & Power Co., Ltd., Mexican Light & Power Co., Ltd., and Barcelona Traction, Light & Power Co., Ltd., the head offices of all of which were located in Toronto. His service to these companies continued until his retirement in 1925, and he therefore played an important part in the inception and development of their hydroelectric properties. On the occasion of the golden wedding anniversary of Mr. and Mrs. Hirt in 1925 they were presented with a gold bowl from these three companies, the inscription on which included a tribute to the skill and technical ability of Mr. Hirt.

Mr. Hirt held many patents in the street-railway field and on gas engines. He had been a member of the A.S.M.E. since 1894, and had belonged to the New York Athletic Club and Lawyers Club prior to his retirement. He is survived by a son, Edward L. Hirt, of Toronto, and by two daughters, Mrs. Sidonia (Hirt) Jackson, of South Orange, N. J., and Mrs. Florence (Hirt) Ellis, of Longmeadow, Mass. His wife, Alice M. (Flammand) Hirt, whom he married in 1875, died in 1934.

PLINY EASTMAN HOLT (1872–1934)

Pliny Eastman Holt, director of the Caterpillar Tractor Company, San Leandro, Calif., died of heart disease at his home in Stockton, Calif., on November 18, 1934. He will be remembered not only for his inventions and improvements in tractors, harvesters, and other farm equipment, and for his service to the city of Stockton as chairman of its Port Commission; but more widely for his developments in the field of ordnance.

In this field Mr. Holt's achievements were threefold. During 1917 he assisted the Ordnance Department of the United States Army in the development of $2^1/_2$-, 5-, and 10-ton artillery tractors, at the Peoria, Ill., plant of the Caterpillar Tractor Company. He was appointed chairman of the Motor Artillery Board and during the remainder of the War was located in Washington, declining a commission and serving without a salary. He completed the designs for a one-man tank, supervised the design of a 150-ton tank, and had begun work on a much larger unit when the War ended. Perhaps his most outstanding achievement was the caterpillar gun mount, the design for which he completed shortly before the close of the War. Subsequently he supervised the construction and testing of a working model. The design provided for the mounting of light- and medium-weight field guns directly on a caterpillar chassis, armed and equipped for military service, and capable of traversing terrain utterly impassable to anything except a track-laying tractor. The design was accepted and the mount has become a standard part of army equipment.

Mr. Holt was born at Loudon, near Concord, N. H., on August 27, 1872. He was the son of William Harrison Holt, who, with his brothers, Charles and Benjamin Holt, started the firm of Holt Brothers in San Francisco in 1869. His mother was Clara Susan (Cate) Holt. His education, secured in the schools of Minneapolis, Minnesota, included three years' special work in mechanical and electrical engineering at the University of Minnesota.

At the age of 23 Mr. Holt took a position with the Carter Brake Company, in Chicago, Ill., as superintendent, and for about a year was engaged in the manufacture of a special street-car brake. In 1896 he became a draftsman for the Holt Manufacturing Company at the Stockton plant. He was made superintendent in 1903, vice-president in 1906. He was also president of the Aurora Engine Company and treasurer of the Houser & Haines Manufacturing Co. He resigned these positions in 1909 to establish the Holt Caterpillar Company at Peoria, but after serving the new company as president and general manager for about a year he was forced to resign because of ill health. In 1912 he became vice-president and general manager of the Holt Manufacturing Company, at Stockton, with which had been consolidated the Holt Caterpillar Company, Aurora Engine Company, and Houser & Haines Manufacturing Co. Ill health again necessitated his retirement as manager in 1916, but he retained the office of vice-president, and since the close of the War had served the company as director.

The inventive ability of Mr. Holt accounted in a large measure for the success of the Holt company and the industrial growth of his community. Nearly all of the agricultural machinery manufactured by the company was designed by him or had met improvement at his hands.

In recent years Mr. Holt had given much of his time to civic activities, and particularly to the Stockton Port Commission, of which he had been chairman since its formation. His careful, first-hand study of the industrial and technical details of the development of the port

were invaluable. He was a member of the Statewide Water Commission and had sponsored the movement for the construction of the Central Valley Water Project. He belonged to the San Francisco and Stockton Chambers of Commerce, was a director of the Stockton Land, Loan and Building Association, and a member of the advisory board of the Bank of America.

Mr. Holt became a member of the A.S.M.E. in 1915 and also belonged to the Society of Automotive Engineers, Society of Agricultural Engineers, Institute of Radio Engineers, American Association for the Advancement of Science, and other engineering and scientific organizations, as well as to the masonic fraternity and to many clubs. He was a director of the Army Ordnance Association and president of the San Francisco post of the Association.

In addition to his widow, Mrs. Florence (Guernsey) Holt, whom he married in 1909, he is survived by three sons and a daughter, Pliny Guernsey Holt, Frank Harrison Holt, Richard Eastman Holt, and Miss Harriet Holt.

BRONSON LOVETT HUESTIS (1890–1934)

Bronson Lovett Huestis, who died on May 29, 1934, was born in Long Island City, N. Y., on June 15, 1890, the son of Robert Sidney and Hanna M. (Isberg) Huestis. He attended the Erasmus Hall High School and Brooklyn Polytechnic Institute, receiving his mechanical engineering degree in 1913. Prior to his graduation, in the summer of 1912, he worked as a patent draftsman.

During the latter part of 1913 and early 1914, Mr. Huestis taught evening classes in machine design at Brooklyn Polytechnic Institute and was employed daytimes by the Stresau Engineering Company, Brooklyn, on the design of special machinery. He was next employed by the International Callophone Corporation, New York, N. Y., first as draftsman on new types of telephone apparatus and later in charge of a small machine shop for developing this apparatus. Later in the year he assisted his father in connection with plumbing and wiring for new residences. After a period of illness he again became connected with Brooklyn Polytechnic Institute, as instructor in machine-shop practice and toolmaker for laboratory apparatus.

At the close of the school year in 1915, Mr. Huestis entered the employ of Westinghouse Church Kerr & Co., New York. He was connected with that company until November, 1917, working as draftsman on sanitation, heating, and ventilation layouts for plants, and testing materials of construction, principally concrete and steel.

In November, 1917, Mr. Huestis became assistant inspector of ordnance matériel for the Ordnance Department of the United States Army, and until April, 1919, was stationed in Connecticut, his duties being at the New Britain Machine Company and the Winchester Repeating Arms Company, New Haven. He was in charge of a division of ballistic tests of small-arms ammunition, and the selection of all Government test specimens at the Winchester company.

In the fall of 1919 he returned to civilian work with the Pratt Engineering & Machine Co., New York, in charge of the mechanical design department, working particularly on chemical plants and equipment. This led to his connection two years later with the Dorr Company, New York, as designer of chemical plants and equipment and the following year as designer of sugar-mill equipment and plants for the Francisco Sugar Company, New York.

Beginning in December, 1922, Mr. Huestis was employed by the American Railway Association, New York, until his death. He was first connected with the Freight Container Bureau, inspecting methods of fruit packing all over the United States, and also carrying on independent research work on freight handling and preparing articles for the technical press on this subject. His book on "Shipping Containers" was published in 1925. In 1923 he became supervisor of the Accident Department of the Bureau of Explosives and subsequently was editor of the "Accident Bulletin" published by the Bureau.

Mr. Huestis became a junior member of the A.S.M.E. in 1913, an associate-member in 1917, and a member in 1923. He was particularly interested in photography, including motion-picture photography. He was unmarried.

EDWARD WALTER JONES (1889–1934)

Edward Walter Jones, vice-president and manager of the Laclede Stoker Company, St. Louis, Mo., died in that city on March 24, 1934. He is survived by his widow, Marguerite (Rickert) Jones, whom he married in 1920, and by two children, Edward R. and Marion F. Jones.

Mr. Jones was born in Chicago, Ill., on March 10, 1889, son of Lewis F. and Margaret (Goss) Jones. After securing a B.S. degree at the University of Illinois in 1911 he became testing engineer for the Commonwealth Edison Company, Chicago, and remained in that position until 1916. He then worked for the American Tar Products Company, Chicago, as fuel and power engineer for about three years. In 1919 he became chief engineer of the Stoker Department of Laclede-Christy Clay Products Company in St. Louis. He supervised the design and development of the Laclede chain-grate, forced-draft stoker. He became vice-president and manager of the Laclede Stoker Company in 1924. He was also active in smoke-abatement work in St. Louis.

Mr. Jones became a member of the A.S.M.E. in 1930 and belonged to the Prairie Club in Chicago.

ALFRED JOHN JUPP (1875–1934)

Alfred John Jupp, a vice-president of The Lunkenheimer Company, Cincinnati, Ohio, died of a cerebral hemorrhage at the Roosevelt Hospital, New York, N. Y., on April 10, 1934, while in the city on business.

Mr. Jupp was born in Cincinnati on June 25, 1875, son of Basil John and Rosina (Kolker) Jupp. He entered the employ of The Lunkenheimer Company at the age of fifteen. In 1896 he was made New York representative and continued in that position until 1913. He then returned to the home office, becoming a vice-president and director of the company.

Mr. Jupp was active in the Manufacturers Standardization Society of the Valve and Fittings Industry and in the Valve and Fittings Institute, and for a number of months prior to his death had contributed much of his time to code work on behalf of the Institute, under the National Industrial Recovery Act. He was a member of the American Committee to the World Power Conference in 1930.

He became an associate of the A.S.M.E. in 1917 and also belonged to the American Society for Testing Materials, Society of Naval Architects and Marine Engineers, American Petroleum Institute, Zoological Society of Cincinnati, American Red Cross Society, masonic fraternity and Elks, and the Engineers', Cincinnati, Queen City, and Cuvier Press Clubs in Cincinnati. He was unmarried and made his home with his sister, Mrs. Sarah J. Rice.

HERBERT EUGENE KAIGHN (1875–1934)

Herbert Eugene Kaighn, consulting engineer, Wilmington, Del., died on March 18, 1934. He was born at Washington, D. C., on August 19, 1875. During his high-school vacations he secured practical experience in the ordnance drafting room at the Navy Yard at Washington, and subsequently took instruction in ordnance construction and electrical engineering there and at the Naval Torpedo Station, Newport, R. I. He also took private lessons in electrical and mechanical engineering.

From 1894 to 1896 Mr. Kaighn was in charge of an electrical course for officers at the Naval Torpedo Station at Newport, and the next year was assistant engineer for the Newport Illuminating Company. In 1898–1899 he was in charge of electrical equipment, marine district, for the New York, New Haven, and Hartford Railroad. From 1900 to 1904 he was again at the Naval Torpedo Station, as ballistic engineer in charge of the powder factory and naval magazine, and the following two years he held a similar position at the Naval Proving Ground, Indian Head, Md. During the early part of 1907 he was at Philadelphia, Pa., in charge of manufacture of powder for the army and navy and the design of machinery for that purpose, and was also inspector of ordnance at the Midvale Steel Works.

Later in 1907 Mr. Kaighn entered the employ of the E. I. du Pont de Nemours Powder Co. at Wilmington as ballistic and experimental engineer in charge of all ballistic and physical tests of powder and the design and improvement of methods and apparatus used in this connection. He remained with the company until 1917, when he established a consulting practice, specializing in designing wiring systems for hospitals and other institutions. Since 1926 he had specialized in radio and acoustic and amplifying systems. He was also consultant for the American Car & Foundry Co. on marine installations.

Mr. Kaighn became a member of the A.S.M.E. in 1913. He also belonged to the American Institute of Electrical Engineers and the United States Naval Institute.

FREDERICK KAREL (1882–1934)

Frederick Karel, appointed car designer by the Board of Transportation of the City of New York in 1927, died on December 8, 1934, after a short illness.

Mr. Karel was born in New York on February 26, 1882, the son of William and Antoinette (Pekarek) Karel. He attended Cooper Insti-

tute from 1902 to 1905 and later took a course in mechanical engineering through the International Correspondence School. Between 1905 and 1920 he was employed nearly all the time by the Central Railroad of New Jersey as locomotive and car designer. Between 1909 and 1916, while serving as senior mechanical draftsman in the office of the superintendent of motive power he instructed apprentices in mechanical drawing and elementary mechanics. He was boiler and construction engineer for the company from 1916 to 1920.

During the next four years Mr. Karel was inspector of locomotives and equipment for the United States Railroad Administration and the Interstate Commerce Commission, and from then until his appointment by the Board of Transportation was chief draftsman in the office of the superintendent of motive power of the Long Island Railroad Company. He did important design work for the Board of Transportation on the 207th Street Shops, Concourse, and Jamaica Yards and in 1911 patented a coupling carry-iron.

Mr. Karel became an associate-member of the A.S.M.E. in 1921 and belonged to the masonic fraternity. He is survived by his widow, Louise (Vanek) Karel, whom he married in 1908, and by a daughter, Mrs. Frances (Karel) Connolly, and two sons, Frederick and Arthur Karel.

EDWARD J. KEARNEY (1868–1934)

Edward J. Kearney, secretary and treasurer of the Kearney & Trecker Corporation, Milwaukee, Wis., died in that city on January 12, 1934, of pneumonia, after having been in failing health for several months.

Mr. Kearney was born at Little Cedar, Mitchell County, Iowa, on April 7, 1868, the son of James Hamilton and Emeline (Smith) Kearney. Left an orphan at the age of 12, he began work as an engine wiper in a roundhouse on the Chicago, Milwaukee & St. Paul R.R. at Egan, S. D. He continued in the employ of the road, working in various capacities, until a year after his graduation, in 1893, from Iowa State College, from which he received the degree of Bachelor of Mechanical Engineering. During the year following his graduation he was located in the West Milwaukee locomotive shops.

In June, 1894, Mr. Kearney took a position as draftsman with the Kempsmith Machine Tool Company, Milwaukee, manufacturers of milling machines. Here he met Theodore Trecker and in 1898, during the Spanish-American war, the two friends decided that if Admiral Dewey were victorious they would start in business together. On May 17, 1898, therefore, a partnership was formed under the name of Kearney & Trecker for the manufacture of milling machines. The business was incorporated under its present name in 1906.

Mr. Kearney became a member of the A.S.M.E. in 1913. He served a five-year term on the Society's standing committee on standardization, terminating as chairman of the committee during 1930. He then became chairman of the Sectional Committee on Allowances and Tolerances for Cylindrical Parts and Limit Gages. After serving the National Machine Tool Builders' Association as vice-president for four years he was elected its president in 1922. He was also secretary and treasurer of the Kearney & Trecker Investment Co., a director of the LeRoi Company, a trustee of Milwaukee Downer College, and one of the original members of the Wisconsin State Board of Vocational Education. During the World War he acted as Director of Sales for the Liberty Loan drives in the Seventh Federal District. He belonged to the Delta Tau Delta fraternity and to a number of clubs in Milwaukee and vicinity.

Mr. Kearney married Ella B. Norton in 1895, and is survived by her and by two daughters, Katherine (Kearney) Carpenter and Alice (Kearney) Fuldner.

WILLIAM THOMAS KEOUGH (1866–1933)

William Thomas Keough, whose death occurred on January 20, 1933, was born in Boston, Mass., on December 22, 1866. He was graduated from the special course in marine engineering and naval architecture at the Massachusetts Institute of Technology in 1888, and after taking post-graduate work and lecturing at the Institute he entered the employ of the Atlantic Works in East Boston. He remained there nine years, the last five as chief engineer.

For the next ten years he carried on work in his own private office as consulting engineer in marine engineering and naval architecture. Then he was elected a member of the Boston School Committee and from 1907 to 1924 held the position of business agent for the school board, establishing modern and efficient business methods. He was appointed a member of the Boston Finance Commission by Governor Allen in 1930 and held that position until his death.

Mr. Keough became a member of the A.S.M.E. in 1900 and belonged also to the Society of Naval Architects and Marine Engineers and the American Society of Marine Engineers. He was a trustee and member of the board of investment of the Home Savings Bank and a director and vice-president of the Enterprise Cooperative Bank. He was a member of the Charitable Irish Society and president of the Catholic Alumni Sodality. He is survived by his widow, who was Rose Butler, of Boston, and by two sons, Paul and John, and a daughter, Katherine.

JAMES WALTER LAWRENCE (1858–1933)

James Walter Lawrence, emeritus professor of the Colorado Agricultural College, Fort Collins, since his retirement in 1917, died on April 3, 1933.

He was born on December 14, 1858, at Toronto, Ontario, Canada, the son of George and Honora (Naughton) Lawrence. His early education was secured in the public schools of Boston, and he subsequently attended the Massachusetts Institute of Technology. After working for several years at the Pacific Mills, Lawrence, Mass., he became a member of the faculty of the Colorado Agricultural College, teaching mechanical engineering and directing the shops and laboratory. He was acting president in 1892 and dean of the faculty from 1907 to 1913. The college conferred an M. E. degree upon him in 1902.

Professor Lawrence also served as president of the Fort Collins School Board from 1902 to 1912. He belonged to the Fort Collins Lodge of Masons and the Colorado School Masters Club, and was treasurer of the Fort Collins Pioneer Society and a member of the St. Lukes Episcopal Church of Fort Collins. He became a member of the A.S.M.E. in 1892.

Professor Lawrence is survived by his widow, Elizabeth (Coy) Lawrence, whom he married in 1890, and by a son, George Coy Lawrence, of Rochester, N. Y.

RUSSELL ELLSWORTH LAWRENCE (1889–1934)

Russell Ellsworth Lawrence, founder of the Lawrence Institute of Technology, located at Highland Park, Mich., died at his home in Redford, Mich., on June 15, 1934. He founded the Institute in 1932 and headed it as president and dean. It was his wish that the school, which in 1933–1934 had an enrolment of more than eight hundred students, be continued as an institution where any student, regardless of financial status, religion, or nationality, may secure engineering training.

He was born in Terre Haute, Ind., on November 4, 1889, a son of Ellsworth Lyman and Mary C. (Holmes) Lawrence. He received a B.S. degree in mechanical engineering from Rose Polytechnic Institute in 1913 and during the next three years was employed in steam-turbine research for the General Electric Company at Lynn, Mass. He then took graduate work at Rose Polytechnic Institute for his master's degree, which he received in 1916. Subsequent work toward his doctor's degree at the University of Cincinnati was interrupted by the World War, and he served in 1918–1919 as a sergeant in the Signal Corps, Aviation Section, United States Army.

From 1919 to 1922 he taught mechanical engineering at the University of Detroit and was acting dean of the College of Engineering. He became dean in 1922 and held that position until he founded the Lawrence Institute of Technology in 1932. He also did a great deal of consulting work while at the University of Detroit.

He became a member of the A.S.M.E. in 1929 and also belonged to the Society of Automotive Engineers and The Franklin Institute, and was a member of the masonic order and Phi Kappa Upsilon fraternity. He is survived by his widow, Mrs. Frieda Lawrence, and by two daughters.

WILLIAM STATES LEE (1872–1934)

William States Lee, inventor of the "Lee pin," an insulator pin used to support large insulators in the transmission of power, and a noted figure in the industrial development of the Carolinas, died at at his home in Charlotte, N. C., on March 24, 1934, of a cerebral hemorrhage. The American Institute of Electrical Engineers, of which he was a past-president, published a tribute in the July, 1934, issue of *Electrical Engineering*, which read, in part:

"Throughout his career, his ability as an engineer, his keen perception of the controlling factors in any situation encountered, and his constant attention to the human values involved made him an outstandingly effective and a highly respected leader of men.

"As president of the American Engineering Council, during the years 1932 and 1933, he worked intensively and enthusiastically to make the engineering profession as helpful as possible in all problems involving engineering and affecting the public welfare."

William States Lee was born at Lancaster, S. C., on January 28, 1872, one of nine children of W. S. and Jenny Lind (Williamson) Lee.

He attended public school at Anderson, S. C., and was graduated as a civil engineer in 1894 from the South Carolina Military Academy at Charleston, now The Citadel. He had held a scholarship in return for which he was obligated to teach for two years in the public schools of the state. In February, 1896, he secured a position with the Greenwood-Anderson & Western Railway. He spent five months in the drafting room, three months as transitman on location, and then was resident engineer on construction until March, 1897. He next became resident engineer with the Anderson Water, Light & Power Co., on a hydroelectric development on the Seneca River, about ten miles from Anderson. His first assignment was to string a transmission line across the river, a task in which others had failed. He was successful and before the close of the year had completed the plant and put it into operation. Following this he served for several months as resident engineer on construction for the Pickens Railway.

From March to November, 1898, Mr. Lee was connected with the United States Engineering Department as assistant engineer on coast defense. He helped with work on forts on Charleston Harbor and assisted in mining the harbor. He was next associated with the Columbus (Ga.) Power Company. From November, 1898, to September, 1901, he was resident engineer for the company, and from then until March, 1903, was its chief engineer. He directed a 8000-hp hydroelectric development and also the construction of a 27,000-spindle fine-yarn mill.

In 1903 Mr. Lee became chief engineer for the Catawba Power Company, Charlotte, a post which brought him into contact with James B. Duke, the tobacco manufacturer. His first task, to dam a river at a plant near Rock Hill, S. C., after three previous attempts to do so had been unsuccessful, was finished in seven months. This was the beginning of the Carolinas superpower system, financially backed by Mr. Duke and designed and built under the direction of Mr. Lee. He served as vice-president and chief engineer of the Southern Power Company, which took over the Catawba Power Company, and later became vice-president and chief engineer of the Duke Power Company, holding this position until his death.

In addition to the huge Carolina system, Mr. Lee designed and built the Isle Maligne station of the Duke-Price Power Company on the Saguenay River, in Quebec, Canada, and was associated with other projects on the Saguenay, part of the Duke system in that section.

So high was Mr. Duke's regard for him that his will provided that Mr. Lee be a trustee of the great Duke foundation of some $80,000,000.

Mr. Lee was president of the W. S. Lee Engineering Corp., of Charlotte, and was retained as consulting engineer on many hydroelectric projects throughout the East and South. He designed and built and for years had operated the Piedmont & Northern Railway, a high-speed, 1,500-volt, direct-current railroad, consisting of about 160 miles of track located in North and South Carolina, an accepted model of electric-railway design, carrying both freight and passengers.

During the World War his advice was sought in connection with plans for naval fortifications along the Atlantic seaboard. He was a director of the American Cyanamid Company, and vice-president and chief engineer of the Great Falls Power Company, Western Carolina Power Company, Wateree Power Company, Catawba Manufacturing & Electric Power Co., Duke-Price Power Company, and the Quebec Development Company, Ltd.

Mr. Lee became a member of the A.S.M.E. in 1907. He also belonged to the Engineering Institute of Canada, the American Society of Civil Engineers, the American Electrochemical Society, the masonic fraternity, and various clubs. In 1929 he was given an honorary degree of Doctor of Science by Davidson College, and he was a trustee of North Carolina State College of Agriculture and Engineering.

He married Miss Mary Martin, of Columbus, Ga., in 1901, and is survived by her and their three children, William States Lee, Jr., Mrs. William H. Williamson, and Martin Lee.

WILLIAM HENRY LEFFINGWELL (1876–1934)

William Henry Leffingwell, president of W. H. Leffingwell, Inc., New York, N. Y., died at his home in Westfield, N. J., on December 19, 1934, of pneumonia.

Mr. Leffingwell was born on June 14, 1876, in Woodstock, Ontario, Canada, the son of American parents, Wendell Phillips and Mary Catherine (Edwards) Leffingwell. After attending high school in Grand Rapids, Mich., he spent ten years as stenographer, clerk, and office manager for different companies. From 1904 to 1910 he was general manager of several large mail-order publications in New York, no longer published. From then until the outbreak of the World War he was in Europe acting as professional office organizer for various firms.

With this varied and practical background Mr. Leffingwell entered more seriously upon his career as a business efficiency expert. He was organization and management engineer at the Chicago Ferrotype Company in 1914–1915, and from then until 1918 was manager of the office-efficiency department of L. V. Estes, Inc., industrial engineers of Chicago. By 1918 he had founded the W. H. Leffingwell Company in Chicago. Two years later the name was changed to The Leffingwell-Ream Company, management engineers, and subsequently the firm was incorporated in New Jersey under its present name.

Mr. Leffingwell believed that for efficient management of an office a highly specialized knowledge was necessary. He applied scientific methods to his task by dividing office management into 93 parts, thus permitting a test rating to be given to each. Then suggestions for improvement were made on the resulting diagnosis. He considered office management inefficient unless each of the 93 parts had the best known practice.

Mr. Leffingwell was the author of a number of books and pamphlets, as well as of articles for scientific societies to which he belonged. His book on "Scientific Office Management," published in 1917, was one of the first on the application of engineering principles to the office. His "Office Management Principles and Practice," published in 1925, is used as a textbook in many schools and colleges. He contributed articles on office management and office appliances to the Encyclopedia Britannica and in 1928 was associate editor of the magazine *System*. He read papers at the Third International Congress of Scientific Management in Rome, 1927, the fourth congress in Paris, 1929, when he was official delegate from the United States, and the fifth congress in Amsterdam, in 1932. He had prepared a paper for the sixth congress to be held in London in 1935.

As aids in carrying out his principles Mr. Leffingwell invented a typewriter measuring scale, a desk and a posture chair for clerks, and a lighting-demonstration device. He enjoyed amateur photography, sculpture, handicraft, and gardening.

Mr. Leffingwell became an associate of the A.S.M.E. in 1921. He was a past-president of the National Office Management Association and was president of the Taylor Society at the time of his death. He was a member of the Society of Industrial Engineers and chairman of its New York Chapter, and belonged to the American Management Association, the Association of Consulting Management Engineers, the Engineers' and City Clubs, New York, the Union League Club, Chicago, and the Sons of the American Revolution.

Mr. Leffingwell married Miss Anna Short, of Chicago, in 1900, and is survived by her and by two daughters, Evelyn E. (Mrs. Joseph W.) Lewis and Dorothy A. (Mrs. Harold F.) Scribner, both of Westfield. His father, two sisters, and a brother, also survive him.

SYLVESTER MAURICE LEONARD (1896–1934)

Sylvester Maurice Leonard, born in Northampton, Mass., on August 16, 1896, died in the Bridgeport (Conn.) Hospital on December 31, 1934. Mr. Leonard, whose parents were James Sylvester and Felicie (Gillet) Leonard, supplemented his grammar school education with study through correspondence schools and by evening work at the New Haven College. He served an apprenticeship in drafting at the Baird Machine Company, Stratford, Conn., and worked as a special machine designer there between 1916 and 1921. Although not continuously employed during the next two and a half years he gained experience in shoe and paper machinery and miscellaneous tool work. During the summer of 1923 he was draftsman at the Continental Paper & Bag Mills, and the following winter worked on wire-drawing and wire machinery for the Waterbury Machine Company. From then until his death he was employed by the Crawford Oven Company, New Haven. He was senior draftsman for about three years and had been chief draftsman since then.

Mr. Leonard became an associate-member of the A.S.M.E. in 1930 and belonged to the masonic fraternity and Odd Fellows. He married Beatrice Hudson in 1925, and is survived by her and their son, Raymond Frederick Leonard.

EDWIN S. LORSCH (1869–1934)

Edwin S. Lorsch, whose death occurred on October 4, 1934, was born in New York, N. Y., on November 19, 1869, a son of Sigmund and Jenny (Simon) Lorsch. He was graduated from the Stevens Institute of Technology in 1891 as a mechanical engineer and during the next two years worked in the drafting, testing, and erecting departments of the Geo. F. Blake Manufacturing Co., manufacturers of pumping machinery, East Cambridge, Mass. He then returned to New York, where he was employed by the Electrical & Mechanical Engineering Co. and in the New York office of the Fort Wayne Electric Company until he became a member of the firm of Sussfeld, Lorsch & Co., in 1896.

Mr. Lorsch was responsible for the organization of the export department of the company in 1899 and for many years was in charge of this department. The firm was reorganized to Sussfeld, Lorsch, and Schimmel on January 1, 1920, and he remained a member of the firm until it was dissolved in 1929.

Mr. Lorsch became a member of the A.S.M.E. in 1903 and belonged to the North Shore Country Club and Harmonie Club of New York. He had made several trips around the world, recorded in a collection of photographs which he took. He was unmarried and is survived by his brother, Henry Lorsch, of New York.

JAMES LYMAN (1862–1934)

James Lyman, consulting engineer, retired, for the firm of Sargent & Lundy, Chicago, Ill., died on March 28, 1934, at Del Monte, Calif., where he had been spending the winter months with his son, Oliver B. Lyman.

Mr. Lyman was born at Middlefield, Conn., on September 1, 1862, the son of David and Catherine (Hart) Lyman. He was graduated from the Sheffield Scientific School of Yale University in 1883 with the degree of Ph.B. and during the summer and fall of that year was foreman of the machine shops of the Metropolitan Manufacturing Company, at Middlefield.

In December, 1883, Mr. Lyman became construction electrician for the Edison Construction Company, with which he remained about six months, assisting in the installation of a number of lighting plants in Pennsylvania and Ohio. During the next two years he was assistant superintendent of the Marr Construction Company, which was organized to take over the construction work for the Edison company. He had entire charge of the installation of plants at a number of places in Pennsylvania, including Johnstown, Wayne, and DuBois, and also gave exhibitions of incandescent lighting.

Resigning from the Marr company in 1886, Mr. Lyman returned to Middlefield as superintendent of the shops and foundry of the Metropolitan Manufacturing Company, in which his family held an interest. During the seven years he spent there the company became part of the American Wringer Company.

In 1893 Mr. Lyman took up post-graduate engineering work at Sibley College, Cornell University, and received an M.E. degree in 1894 and M.M.E. degree in 1895. He then entered the employ of the General Electric Company at Schenectady and as assistant to Dr. Charles P. Steinmetz conducted special experimental testing and designed electric apparatus. Later he was transferred to the power and mining department and in 1899 was appointed assistant engineer of the Chicago branch of the company. Three years later he became district engineer and continued in that capacity until 1911.

From 1911 until his death Mr. Lyman was associated with the firm of Sargent and Lundy, Inc., as electrical engineer, member of the firm, vice-president (1926–1930), and as consultant retired since 1930.

Mr. Lyman became a member of the A.S.M.E. in 1899, was a Fellow of the American Institute of Electrical Engineers, and belonged also to the American Electrochemical Society, Western Society of Engineers, and the Institution of Electrical Engineers. He was a member of the University and Union League Clubs, Chicago, and several clubs in Evanston, where he made his home for more than thirty years.

Mr. Lyman married Miss Anna J. Bridgman, of Boston, Mass., in 1891.

WALTER ARNOLD LYNCH (1902–1934)

Walter Arnold Lynch, who died of typhoid fever and meningitis at Carrizo Springs, Texas, on September 23, 1934, was born in Philadelphia, Pa., on August 16, 1902, the son of Edmund Wright and Alice (Clayton) Lynch. He attended high school in Darby, Pa., and received a B.S. degree in mechanical engineering at Drexel Institute, Philadelphia, in 1926.

Between the time of his graduation and 1929, Mr. Lynch engaged in farming in Carrizo Springs and was employed at different periods by the Atlantic Refining Company, Philadelphia, in the conversion of a ship from steam to Diesel-electric drive; by the Mayhue Lumber Company in the installation of deep-well pumps; and by the Central Power & Light Co. in the installation of refrigeration machinery.

In 1929 he secured employment as a junior hydraulic engineer with the Texas State Board of Water Engineers in an underground water survey, and in 1933 began similar work with the United States Geological Survey, with which he was connected at the time of his death.

Mr. Lynch had devised a portable device for testing for salt water in deep wells and for determining the position of the salt-water leaks. He also designed and constructed a wind-driven generator for charging a storage battery on his farm, and had constructed many models for airplanes. Radio was perhaps his principal hobby; he had constructed many sets and was a licensed amateur sender.

He served as chairman of the A.S.M.E. Student Branch at Drexel Institute in 1926 and became a junior member of the Society upon his graduation. He was also president of the Phoenix Club at the Institute in 1926, and served as a sergeant in the Headquarters Battery of the 108th Field Artillery of the Pennsylvania National Guard from 1921 to 1925. He was unmarried.

NELSON THEODORE MANN (1899–1934)

Nelson Theodore Mann, who had been employed as a research analyst since 1927 by Scudder, Stevens & Clark, New York, N. Y., died on February 17, 1934, of pneumonia. He is survived by his widow, Beatrice Smith Mann, whom he married in 1928, and by a son, Timothy Charles Nelson Mann.

Mr. Mann was born at Hawley, Mass., on November 10, 1899, attended the Medford (Mass.) High School, and secured an S.B. degree in Mechanical Engineering at the Massachusetts Institute of Technology in 1923. He spent one vacation as rodman for the Scully Construction Company, Boston, and several with the New York Central Railroad, working as pipefitter's helper and in other capacities in the Electrical Department. He served nine months in the United States Marine Corps during 1918.

After his graduation Mr. Mann served a special apprenticeship with the Westinghouse Air Brake Company, Wilmerding, Pa., and worked for that company as assistant to the mechanical engineer and as commercial engineer for a time prior to his connection with Scudder, Stevens & Clark. His work for the latter company had been mainly on machinery, tin containers, railroad equipment, and nonferrous metal mining, analyzing their investment possibilities.

Mr. Mann became a junior member of the A.S.M.E. in 1924, and belonged to the New Rochelle Tennis Club and the Huguenot Yacht Club.

NORMAN ISAAC MARSHALL (1865–1933)

Norman Isaac Marshall, who died in New York, N. Y., on March 28, 1933, was born at Hampstead, N. H., on December 6, 1865, the son of Isaac Hull and Julia (Bement) Marshall. He prepared for college at the Bromfield Academy, Harvard, Mass., and was graduated from Worcester Polytechnic Institute in 1886 with a B.S. degree in mechanical engineering.

For several years following his graduation Mr. Marshall was employed by the Marr Construction Company, assisting in the erection of central-station light and power plants in Pennsylvania, Ohio, and Wisconsin. He next had charge of similar work for the Westinghouse Electric & Manufacturing Co. in Texas, later becoming eastern manager for the company, with headquarters in Boston. From 1901 to 1903 he was designer of machinery and fittings and superintendent of shops for the Star Electric Company, Philadelphia, Pa., and during the next two years manager of the Iona Manufacturing Company and Anchor Electric Company, Boston.

In 1905 Mr. Marshall went into business for himself as owner and manager of the Marshall Electric Company, Boston. Subsequently he sold this business and formed the Elastoid Fibre Company, in West Newton, Mass., later located in Waltham, Mass. He was president and treasurer of the company until its sale in 1927, since when he had not been active in business except for consulting work.

Mr. Marshall held many patents in the electrical and mechanical fields. He was one of the first to manufacture sockets, switches, fuses, etc., and patented the first automatic machinery for the manufacture of incandescent bulbs. From 1916 to 1918 he was connected with the science and research department of the United States War Department, and jointly with Dr. Louis Bell invented and perfected the ultra-violet signalling apparatus for use in the Signal Corps during the World War.

Mr. Marshall served as a lieutenant in the naval militia during the Spanish-American War.

Since 1927 Mr. Marshall had spent a great deal of his time in travel. He was an excellent shot with the rifle and had hunted big game in Alaska, China, and India. He wrote several articles on hunting for publication, and had also written some short stories. He greatly enjoyed walking and mountain climbing and was an able botanist.

Mr. Marshall became a member of the A.S.M.E. in 1910 and also belonged to the American Institute of Electrical Engineers, Boston University Club, Lake Placid Club, and Appalachian Mountain Club.

He married Miss Hortense Carver in 1898 and is survived by her and by three children, Thomas C. Marshall, of Fairfield, Conn., Mrs. M. H. Hull, Still River, Mass., and Mrs. C. V. Russell, Boston.

WILLIAM CROSBY MARSHALL (1870–1934)

William Crosby Marshall, consulting mechanical and civil engineer, and former professor of machine design at the Sheffield Scientific School, Yale University, died at the Gaylord Farm Sanitorium, Wallingford, Conn., on February 1, 1934. Mr. Marshall, a descendant of Robert Treat, an early governor of Connecticut and founder of Newark, N. J., retired in 1929 and since that time had made his home at Milford, Conn. He was born at Avon, Conn., on September 21, 1870, the son of Henry G. and Marietta (Crosby) Marshall.

After being graduated from Yale University in 1890, with a Ph.B. degree, Mr. Marshall worked for three years as a draftsman and structural engineer for the Berlin Iron Bridge Company, Berlin, Conn. He then returned to Yale to study for a mechanical engineering degree, which was conferred upon him by Sheffield Scientific School in 1894. He remained at the school, first as instructor and later as professor in machine design, until 1913. He was given a civil engineering degree there in 1900. He studied at the École des Beaux Arts and École du Genie Maritime, in Paris, in 1907–1908.

For a year after he left Yale in 1913 Mr. Marshall was chief engineer for the Federal Sugar Refining Company, New York, N. Y. He then became research engineer at the Union Metallic Cartridge Company, Bridgeport Works of the Remington Arms Company. During the early part of 1917 he was connected with the Holt Manufacturing Company, Peoria, Ill., as mechanical engineer, then served during the World War as captain in the Ordnance Department, U. S. A., in charge of twenty-five plants. He also served as trade commissioner to Italy for the U. S. Bureau of Foreign and Domestic Commerce in 1919.

Returning to civil life, Mr. Marshall was engaged as mechanical engineer for the National Spun Silk Company, New York, and carried on consulting work in that city on machine designing, testing, graphical charting, and materials handling and traveled in England inspecting various plants for new and comparative material. In 1921–1922 he was chief enginer for the U. S. Hoffman Machinery Corp., New York; 1922–1923, mechanical engineer for The Trexler Co. of America, Wilmington, Del.; 1923–1924, maintenance and plant engineer, Richard Hellman, Inc., Long Island City, N. Y.; 1924–1925, director, Yale Graduate Placement Bureau, Yale Club, New York; 1925–1926, instructor in machine drawing, Pratt Institute, Brooklyn, N. Y. From then until his retirement he was an associate editor of The Engineering Index.

While living in Milford, Mr. Marshall served as town-planning and building inspector there. He was the author of a textbook on "Descriptive Geometry," published in 1910, "Machine Drawing and Design," 1912, and "Graphical Methods," 1921, as well as of many magazine articles.

Mr. Marshall became a junior member of the A.S.M.E. in 1901 and a member in 1909. He also belonged to the Sigma Xi fraternity, Yale Engineering Association, and Yale Club, New York. He is survived by his widow, Mrs. Genevieve Holbrook Marshall, and by a son, John.

ROYAL MATTICE (1878–1934)

Royal Mattice, president of the Mattice Engineering Company, Philadelphia, Pa., died at his home in that city on February 18, 1934, and was buried with military honors in the Arlington National Cemetery. He was born at Birmingham, Ala., on July 27, 1878. At the age of 13 he began work as a water boy at the Carnegie Steel Company. He served an apprenticeship as a patternmaker and for six years took evening courses, specializing in metallurgy. In 1897 he was employed as assistant manager and metallurgical engineer for the Kokomo (Ind.) Nail & Brad Co., and after serving in the Spanish-American War in 1898, returned to that company in the position of general manager in charge of sales and production.

From 1900 to 1920, with the exception of periods of service on the Mexican Border and in the World War, Colonel Mattice was metallurgical engineer for the American Steel & Wire Co. He was division sales manager for the company at Chicago, Ill., Oklahoma City, Okla., Wichita, Kan., and Cincinnati, Ohio. Although his connection with this company, as well as with the Kokomo Nail & Brad Co., was officially in the sales department, he had practically nothing to do with selling except in a technical way. He was chiefly concerned with improvements in the firm's products and production methods.

After the World War, Colonel Mattice established the Mattice Engineering Company, devoted almost entirely to electric welding of cast iron.

Colonel Mattice became a member of the A.S.M.E. in 1924, and also belonged to The Franklin Institute, the Military Order of Foreign Wars, the Engineers Club in Philadelphia, and the masonic order. He was president of the Welding Contractors' Association of Greater Philadelphia and Camden.

He is survived by a son, Royal Mattice, Jr., and a daughter, Nancy Ellen.

DABNEY HERNDON MAURY (1863–1933)

Dabney Herndon Maury, retired consulting engineer, chiefly in the water-supply field, died at his home in Chevy Chase, Md., on May 11, 1933. Mr. Maury was born in Vicksburg, Miss., on March 9, 1863, the son of Dabney Herndon and Nannie Rose (Mason) Maury. His early education was secured in private schools in New Orleans, La., and Richmond, Va. He then attended the Virginia Military Institute, at Lexington, Va., from which he was graduated in 1882. Two years later he secured his M.E. degree from Stevens Institute of Technology, Hoboken, N. J.

Mr. Maury worked during his summer vacations while attending college. In 1881 he was rodman on surveys for a railroad bridge across the Ohio River at Point Pleasant, W. Va. The following summer he was chief of a party making preliminary surveys for the Brighthope Railway, near Richmond, Va., and draftsman at Colt's Armory, Hartford, Conn. During the summer of 1883 he was draftsman for locomotive works in Richmond and in 1884 assistant to Prof. Robert H. Thurston in the mechanical laboratory at Stevens Institute.

In October, 1884, Mr. Maury went to Texas as engineer for the Grand Belt Copper Company, of New York, a connection he retained until the summer of the following year. He then became principal assistant engineer on location and construction for the Ft. Worth and New Orleans Railway. From March to November, 1886, he was engaged in making land surveys for the Southern Pacific Railway Company in western Texas.

From Texas Mr. Maury went to Colombia, S.A., where his father, General Dabney H. Maury, was stationed as Envoy Extraordinary and Minister Plenipotentiary of the United States to Colombia. He served as private secretary to his father until March, 1887, when he became general engineer for Green & Garcia, mine agents in Colombia. He remained in their employ until May, 1890, serving part of the time as general manager of the Tolima, Organoz, Socorro, and other gold and silver mines, erecting machinery and building dams and ditches for the Silencio, Tetuan, and Colon mines, and making surveys, plans, and estimates for other mines in Colombia.

During the remainder of that year Mr. Maury traveled in Great Britain and the United States. He returned to Colombia as general manager for the Saldana Syndicate, Ltd., of Liverpool, England, and until the winter of 1892 had charge of all of its interests there, designing and constructing dams and ditches and directing the operation of mines. He was also general manager for the Anape Syndicate, Ltd.

After his return to the United States in 1892 Mr. Maury took post-graduate work at Stevens Institute through the winter. He then became engineer and superintendent for the Peoria (Ill.) Water Company and later continued in the same capacities with the reorganized Peoria Water Works Company, remaining with the company until August, 1912. He designed and constructed a number of new pumping plants, developed improvements in operation, and designed sewer systems for suburban districts.

During the latter part of this period he had also conducted a private practice as a consulting engineer, and in August, 1912, he opened his own office in Chicago. He was located there until the United States entered the World War in 1917, and again from July, 1919, until his retirement in 1927, during the last four years as senior partner in the firm of Maury and Gordon. His advice on water supplies was sought by municipalities in many parts of the country.

In May, 1917, Mr. Maury put aside his private practice to serve as advisory engineer on water supply for the Construction Division of the United States Army. He was in charge of the design of water systems for all projects of the Division in the United States and possessions. He was also borrowed by the Marine Corps for its water supplies at Paris Island and Quantico, and by the Navy for supplies at Pearl Harbor and other points in Hawaii. He was specially assigned by the War Industries Board to develop water supplies for the army, navy, shipping board, and housing corporation in the Hampton Roads district. He was given the rank of commanding major in the Engineers Reserve Corps in 1917 and of lieutenant-colonel in the Quartermaster Corps in 1918. He was honorably discharged in May, 1919.

In 1920 Mr. Maury won the James Laurie Prize of the American Society of Civil Engineers for a paper on "Water Supplies for the Cantonments, Camps, and Other Activities Built by the Construction Division of the Army." He contributed other papers to the technical press and to the societies to which he belonged. He held several patents on pumping machinery and wells.

Mr. Maury became a member of the A.S.M.E. in 1890. He was past-president of the American Water Works Association, Illinois

Society of Engineers, and Illinois Water Supply Association. He was also a member of the American Society of Civil Engineers, American Institute of Consulting Engineers, Western Society of Engineers, and New England Water Works Association, as well as of Kappa Alpha (Southern) and Tau Beta Pi, honorary engineering fraternity, and a number of clubs.

Mr. Maury married Mary McCaw, of Richmond, in 1893, and is survived by her and their son, Dabney Herndon Maury, Jr.

RICHARD JUSTIN McCARTY (1851–1934)

Richard Justin McCarty was born on March 12, 1851, at Clarksburg, W. Va., the son of Joseph and Ann (McCally) McCarty. He attended Soulé University, Chapel Hill, Texas, from 1862 to 1869 and then studied for two years at the University of Virginia, being graduated in pure mathematics in 1871. Subsequently he took courses in applied mathematics and civil engineering there, completing his work in 1875.

Following a short period as foreman for the erection of a rolling mill at Rosedale, Kan., he was employed by the Kansas Rolling Mill Company in various capacities until 1879. He then worked in the engineering, accounting, and operating departments of steam railroads until 1888, when he became chief engineer of the Metropolitan Street Railway Company at Kansas City, Mo. He held this position until 1895, having charge of designing and constructing extensions to the cable lines and power plants and the maintenance and operation of all of the company's property. After he left the regular service of the company in 1895 he was recalled to make changes in another of its lines and was also employed in 1895 and 1896 by the Kansas City Stock Yards Company to design a pumping and lighting plant and to handle construction work.

In 1897 Mr. McCarty became auditor for the Kansas City, Pittsburg & Gulf R.R., which later became the Kansas City Southern Railway. In 1903 he resigned to devote two years to scientific study, then returned to his former position. Later he was made vice-president of the company and chairman of its valuation committee. He retired permanently in 1918 and spent his remaining years in scientific and philosophical study and writings. His work on "Valuation of Railroad Property" was published in 1915; "Elements of Plane Trigonometry, 1920; Wisdom and Purpose, 1922; Work and Play: An Autobiography, 1925; "Essays in Poetry," written jointly with his wife, 1930; and Vital Evolution, 1933.

He died in Kansas City of heart disease on June 16, 1934. He is survived by his widow, the former Miss Mary Louise Allen, whom he married in 1877, and by three sons, Allen, Richard J., and Charles E. McCarty.

Mr. McCarty became a member of the A.S.M.E. in 1901 and also belonged to the American Society of Civil Engineers, Sons of the American Revolution, Royal Societies Club, London, and the Zeta Psi fraternity.

JOHN McGEORGE (1852–1933)

John McGeorge, honorary member of the Cleveland Engineering Society, was instantly killed on the evening of February 27, 1933, when struck by an automobile while walking near his home in Cleveland. Mr. McGeorge had been a member of the A.S.M.E. since 1891.

He was born in Manchester, England, on May 2, 1852, Leaving school when about fourteen years of age, he served a seven-year apprenticeship with Emmerson Murgatroyd & Co., Stockport, England, working in both the shops and drawing office. He also studied evenings at the Mechanic's Institution of Stockport and later, while employed in Manchester, took evening courses at the Owens College there.

After completing his apprenticeship, Mr. McGeorge worked successively as draftsman with Galloway & Sons, engineer with the Boiler Insurance Company, and draftsman and manager of Deakin Parker & Co., all of Manchester. He was connected with the Deakin company for about seven years and during this period invented a drum or shaft governor. Later he engaged in drafting for companies in Nottingham, Grantham, and Guilford, England.

In 1884 Mr. McGeorge came to the United States. During his early years here he constructed special machinery for the Bellaire Stamping Company, Bellaire, Ohio, including machinery of his own design for making the Mason fruit jar cap. Subsequently he was employed by the Pittsburgh Iron & Steel Engineering Co., Pittsburgh, Pa., as chief draftsman.

In 1892 he became associated with Samuel T. Wellman, of the Wellman-Seaver Engineering Company (now the Wellman Engineering Company), and served as chief engineer of the company from 1896 to 1903. With Mr. Wellman he developed the rolling open-hearth furnace, the open-hearth charging machine, and other equipment which meant much to the development of the steel industry. Since 1903 his work had been largely in the consulting field. He also invented an electric factory truck in wide use as a labor-saving device in the handling of materials.

Mr. McGeorge was elected to membership in the Cleveland Engineering Society in 1896 and honorary membership was conferred upon him in 1931. He was also a life member of the Society of Automotive Engineers.

DANIEL GEORGE McMILLAN (1871–1934)

Daniel George McMillan, engineer for the Jersey Central Power & Light Co., South Amboy and Sayreville, N. J., died in Elizabeth, N. J., on May 30, 1934. He is survived by his widow, Eleanor T. (Christie) McMillan, whom he married in 1907, and by a daughter, Mildred Eleanor.

Mr. McMillan was born at St. Louis de Gonzague, Quebec, Canada, on May 14, 1871, the son of Neil and Flora (Menish) McMillan. He secured his early education in Montreal schools, served an apprenticeship at the Royal Electric Company there, and spent three years in the mechanical and electrical laboratories of McGill University. For five years he served as chief engineer of the Canadian Singer Factory at St. Johns, Quebec, then came to the United States as assistant chief engineer of the Singer Manufacturing Company at Elizabethport, N. J. He remained with this company for nearly ten years, being its chief engineer the latter part of the time. Subsequently he was superintendent of power for the Willys Corporation, Elizabeth, and maintenance engineer of the Elizabeth General Hospital, prior to his connection with the Jersey Central Power & Light Co.

Mr. McMillan became an associate-member of the A.S.M.E. in 1917. He was very active in the masonic fraternity, in which he had held a number of important posts, and was a member of Everymans Bible Class at the Third Presbyterian Church in Elizabeth.

HIRAM LEES MELLOR (1865–1933)

Hiram Lees Mellor, owner of the Lawrence Pump & Engine Co., Lawrence, Mass., a member of the A.S.M.E. since 1908, died on January 5, 1933. He was born at Sandwich, N. H., on December 2, 1865, but moved with his parents to Lawrence at an early age and was educated in the public schools there. He served an apprenticeship as a machinist and draftsman with S. S. & W. O. Webber, Lawrence, manufacturers of textile machinery, printing presses, shop tools and jigs, and attended night school, thus fitting himself for a position as draftsman with the Lawrence Machine Company. He spent four years in that work, then became superintendent of the company, which manufactured centrifugal pumping machinery, steam engines, and related equipment. About 1904, he started in business for himself, founding the Lawrence Pump & Engine Co., which he continued to manage until his death.

He had served as treasurer and deacon of the United Congregational Church in Lawrence. He was married, and is survived by his widow and a daughter, Mrs. J. L. Dean, whose husband now manages the business.

EDGAR HAMILTON MERRICK (1886–1934)

Edgar Hamilton Merrick, whose death occurred in Cleveland, Ohio, on January 21, 1934, was born at Gananoque, Ontario, Canada, on May 4, 1886, son of Dr. Edgar H. and Sara E. (Carpenter) Merrick. He was graduated from Sibley College, Cornell University, in 1908, with an M.E. degree, and during the next six years was machinery salesman for Hill, Clarke & Co., Chicago, Ill. He then became secretary and sales manager for the Perkins Windmill & Engine Co., Mishawaka, Ind., where he remained for three years.

From 1917 to 1927 Mr. Merrick was connected with Manning, Maxwell & Moore, Inc., New York, N. Y., as district manager of their Cincinnati office for seven years and of the Cleveland office for three years. He engaged in machinery business for himself during the next year, then became sales engineer for Dravo-Doyle Company, of Pittsburgh, Pa., handling the products of the Shepard-Niles Crane & Hoist Corp. Since 1931 he had been in business for himself as sales engineer, in Cleveland.

Mr. Merrick became an associate of the A.S.M.E. in 1932, and was a 32d degree Mason. He is survived by his widow, Florence C. (Earle) Merrick, whom he married in 1917, and by two children, Frances Chapin and Richard Henry Merrick.

VLADIMIR VICTOR MESSER (1877–1932)

Vladimir Victor Messer, who died in 1932, on or about May 2, was born in St. Petersburg, Russia, on December 2, 1877. He attended

the naval academy of the Imperial Russian Navy and was commissioned a lieutenant in 1896. His first work in the United States was as chief draftsman for the Los Angeles Gas & Electric Co., in 1906. The following year he became consulting engineer for the Sweetland Filter Press Company and Brubaker Bros., in San Dimas, Calif., and subsequently served the Commercial Gas Engine Company, of Los Angeles, in connection with irrigation equipment. In 1912 he engaged in the design of the steel structure for the Canadian Pacific Railway terminal at Vancouver, as an employee of Coughlan & Sons, of that city. He then returned to Los Angeles, where he became structural engineer for Fitzhugh & Fitzhugh, architects. He subsequently held a similar position with Lescher & Kibbey, architects, in Phoenix, Ariz.

About the year 1915 Mr. Messer went to New York, where he was associated with Percival R. Moses and Frederick Pope as consulting engineer. During the World War, in 1918-1919, he was chief engineer for the Newport Chemical Works and Newport Hydro-Carbon Company, Milwaukee, Wis., and from then until 1924 again associated with Moses and Pope. At that time he formed the V. V. Messer Manufacturing Co., located first in Long Island City, and later in New York.

Much of his time during the latter part of his life was spent in work on inventions and on special missions to other countries, including the building of a steel plant and railroad in Russia, and zinc plant in South America, and in connection with the cork industry in Europe. Other special connections were with the Goodyear Tire & Rubber Co., in Akron, Ohio, the National Analine Company, for which he built an indigo plant in Buffalo, N. Y., and the Miller Rubber Company, Inc., New York.

Mr. Messer became a member of the A.S.M.E. in 1916 and belonged to the Chemists' Club, New York. He is survived by his widow, Edithe Maxham Messer, whom he married in 1917. Their only son, Paul, died in October, 1933.

HAROLD R. MILLER (1906–1934)

Harold R. Miller was born at Norwalk, Ohio, on July 3, 1906. He secured his early education there and was graduated from the Ohio State University in 1928 with a B.S. degree in mechanical engineering. He immediately entered the employ of the Philip Carey Manufacturing Company, at Lockland, Ohio, and at the time of his death on March 31, 1934, was assistant superintendent of power.

Mr. Miller became a junior member of the A.S.M.E. in 1933 and belonged to the Theta Xi and Tau Beta Pi fraternities. He is survived by his widow, Mary (Ross) Miller, whom he married in 1926, and by a daughter, Marjorie Anne Miller.

JAMES MILNE (1865–1934)

James Milne, chief electrical and mechanical engineer in the Department of Works of the City of Toronto, Ontario, Canada, in charge of water supply, for about twenty years, died in that city on May 21, 1934.

Mr. Milne was born on January 29, 1865, at Woodside, Aberdeen, Scotland, the son of William and Jessie (Hird) Milne. He secured his technical training at Robert Gordon's College, Aberdeen, and as an apprentice with C. Davidson & Sons and Abernethy & Co., Aberdeen. He then went to Montreal, Canada, where he worked for a time in a machine shop and later secured employment with the Edison General Electric Company. He spent three years there and during the next six years was superintendent of the Incandescent Electric Light Company at Toronto. For two years he taught machine construction and drawing, applied mechanics, and steam engineering at the Toronto Technical School, then spent eight years as general manager and chief engineer of the Underfeed Stoker Company, Toronto, at the same time carrying on a consulting engineering practice. Early in February, 1906, he became general superintendent of the British Columbia Electric Railway Company, Vancouver, operating street railways, light, and power in Vancouver, Victoria, and New Westminister, B.C. From 1910 to 1913 he was assistant manager of the Windsor Salt Works, Windsor, Ontario, and engaged in consulting work. He began his long connection with the City of Toronto in 1913.

Mr. Milne became a member of the A.S.M.E. in 1907. He also belonged to the American Institute of Electrical Engineers, Engineering Institute of Canada, and the Association of Professional Engineers of Ontario, and was a member of the Engineers and Parkdale Canoe Clubs in Toronto and of the masonic fraternity. He was a contributor to technical publications and a member of the Toronto Symphony Orchestra, in which he played the contra-bassoon.

Mr. Milne had been twice married. His first wife died in 1899 and his second in 1929. He is survived by a son, James C. R. Milne, and by a married daughter.

GUY KÖCHLING MITCHELL (1878–1934)

Guy Kochling Mitchell, formerly president and proprietor of the Standard Electric Machinery Company and partner in the Standard Appraisal Company, both of Baltimore, Md., died of pneumonia at his home in that city on January 15, 1934.

Mr. Mitchell was born in Baltimore on July 5, 1878. He attended the Baltimore Polytechnic Institute and in 1899 began work for the United Electric Light & Power Co. as draftsman for the department handling equipment and operation. He spent six years in that position, then in 1902, after a few months at Embreeville, Tenn., installing an electric plant and transmission line for the Embree Iron Company, he returned to Baltimore to do estimating and layout work in the distribution department of the United Electric Light & Power Co.

In 1906 Mr. Mitchell became manager of the electrical department of Crook Horner Company, Baltimore. When this company dissolved the following year he purchased the electrical and elevator departments and established the Standard Electric & Elevator Co. operating it as a manufacturing and contracting concern. The name of the company was changed to the Standard Electric Machinery Company in 1921.

During the early days of the company it specialized in replacing steam drives by electric-motor drives, and under the direction of Mr. Mitchell installations varying from 50 hp to 1000 hp were made in plants in Baltimore and vicinity. At the same time he carried on the development of electric elevators, especially the single-phase, and a new line of dumbwaiters, hoists, and cranes. During the World War he devised an automatic system for reconnecting and rerating motors and generators of all makes and types.

Mr. Mitchell became an associate of the A.S.M.E. in 1913 and a member six years later. He was also a member of the American Institute of Electrical Engineers. He is survived by his widow, Katherine V. Mitchell.

ROBERT LEVIS MITCHELL, JR. (1911–1934)

Robert Levis Mitchell, Jr., son of Doctor and Mrs. Robert L. Mitchell, of Baltimore, Md., died at Perth Amboy, N. J., on December 28, 1934. He had been a member of the engineering staff of the National Lead Company at Perth Amboy since early in July.

Mr. Mitchell was born in Baltimore on September 21, 1911. He attended the Friends School there and secured his degree in mechanical engineering from Johns Hopkins University with the class of 1934. He was a student member of the A.S.M.E. at the university and became a junior member of the Society upon graduation. He was also a member of the university amateur theatrical group, the Barnstormers, and of Alpha Tau Omega fraternity.

Surviving Mr. Mitchell, in addition to his parents, are a sister, Miss Nancy Mitchell, and a brother, William.

LESTER A. MORELAND (1874–1934)

Lester A. Moreland, former production engineer for the Worthington Pump & Machinery Corp., Harrison, N. J., died on March 21, 1934, after an illness of more than a year. Mr. Moreland was born at Skaneateles, N. Y., on December 11, 1874, the son of Lewis P. and Harriet (Daniels) Moreland. After leaving public school he began to learn the machinist's trade with the Straight Line Engine Company, of Syracuse, N. Y., but soon decided that he would prefer toolmaking. He therefore transferred to the E. E. Stearns Co., Syracuse, and there and in other shops learned the trade.

In 1898 he entered the employ of the Oliver Typewriter Company, at Woodstock, Ill., as a toolmaker, and soon was placed in charge of of the department. As the company had no drafting or designing department, he made what drawings were required. After six years with the company, he became foreman of the tool designing, experimental, and toolmaking departments of the Illinois Sewing Machine Company, at Rockford. Later he was made assistant superintendent in full charge of all departments, including the foundry and patternmaking.

In 1908 Mr. Moreland went to Providence, R. I., taking charge of the machine department of the American Locomotive Automobile Company, and later of its toolroom and grinding department. The following year he became tool supervisor of the Warner Gear Company, Muncie, Ind., with which he was connected until 1915. One of the special machines which he designed for the company was an automatic machine for inserting corks into clutch plates for automobile clutches.

After leaving the Warner company, Mr. Moreland spent a year with the Remington Arms Company, at Eddystone, Pa., as assistant superintendent of the tool department, and two years with the British Munitions Company, of Montreal, Canada, as master mechanic in the fuse loading and assembling plant.

In 1918, when the World War was nearly over, Mr. Moreland went to Trenton, N. J., to become chief engineer and designer of the John E. Thropp's Sons Co. Among the designs made for the company by him were a special machine for engraving non-skid designs in tire molds, a fabric-spooling machine for taking the fabric from the bias cutter, a special banding machine, and a collapsible core and chuck. He had been with the company about ten years when he became production engineer for the Worthington Pump & Machinery Corp.

Mr. Moreland became a member of the A.S.M.E. in 1923, and belonged to the Engineers Club in Trenton and to the masonic fraternity. He is survived by his widow, Emma J. (Chappel) Moreland, whom he married in 1899, and by a son, Lester D. Moreland.

GEORGE WILLIAM MORETON (1860–1934)

George William Moreton, who died in Wilmington, Del., on March 2, 1934, after a year's illness, was English by birth. The son of John and Annie (Smith) Moreton, he was born on July 23, 1860, at Minshull Vernon, Cheshire, England. He secured his early education at public schools in Crewe and Birmingham and later was a science and art student at the Crewe Mechanics Institute, and winner of a Whitworth Scholarship in 1885.

Mr. Moreton began work as an office boy for the London and Northwestern Railway in Birmingham at the age of fourteen, and three years later was transferred to its shops at Crewe as an apprentice. He completed his training in 1881 and continued to work for the company as machinist and draftsman until he came to the United States.

Mr. Moreton's first employment in this country was with the Pond Machine Tool Company, Plainfield, N. J., with which he was connected as draftsman from 1888 to 1892. He then engaged in similar work for the Betts Machine Company, of Wilmington, of which he became general superintendent in 1898, and later served also as secretary and vice-president. When the company was sold, during the World War, becoming the Betts Shops of E. I. duPont de Nemours & Co., Mr. Moreton was retained as superintendent and consulting engineer for several years. Since then he had been associated with the Consolidated Machine Tool Corporation of Rochester, N. Y., as consulting engineer and manager of the Hilles & Jones plant in Wilmington.

Mr. Moreton became a member of the A.S.M.E. in 1909. He also belonged to the Wilmington Country and Wilmington Whist Clubs and the Whitworth Society of England. He was a director of the Artisans Saving Fund, Wilmington, and a member of Trinity Church there. He is survived by his widow, Hilda W. (Brotherhood) Moreton, whom he married in 1906, and by two children, John B. and Winnifred Moreton.

THOMAS MORRIN (1853–1934)

Thomas Morrin, son of John and Mary Morrin, was born at Waterloo, N. Y., on August 6, 1853, and died at his home in San Francisco, Calif., on August 1, 1934. He attended public school in Waterloo and at the age of fifteen began an apprenticeship at the Island Works in Seneca Falls, N. Y., where steam fire engines were manufactured. After the completion of his apprenticeship in 1871 he worked for fifteen years as a machinist, being employed in Bay City, Mich., and San Francisco and Marysville, Calif.

In 1886 Mr. Morrin became master mechanic for the Carson & Tahoe Lumber & Co., at Glenbrook, Nev., where he had full charge of railroad, steamboat, and sawmill machinery. Two years later he accepted a similar position with the Western Beet Sugar Company in Watsonville, Calif., where he remained until July, 1890. During the next two years he was foreman in charge of the construction of a machine shop, then became master mechanic and chief engineer for the D. O. Mills properties in and near San Francisco, a position which he held until 1908.

Mr. Morrin also practiced independently as a consulting mechanical engineer from 1904 until 1928, when he practically retired, though he continued to act as supervising engineer for the Phelan estate, San Francisco, until the date of his death.

Mr. Morrin became a member of the A.S.M.E. in 1897 and belonged to the Engineers Club in San Francisco. He was especially interested in fine cabinet work in mahogany and walnut, and took walking trips into the Sierra Nevada Mountains. He is survived by a daughter, Miss Mary Irene Morrin. His wife, Irene (Hoyt) Morrin, died in 1916.

VICTOR EMANUEL MUNCY (1860–1934)

Victor Emanuel Muncy, whose death occurred at Ashland, Ky., on October 26, 1934, was born at Louisa, Ky., on December 1, 1860, the son of Samuel K. and Teresa (Carter) Muncy. After teaching at the Academy of the State (Agricultural and Mechanical) College of Kentucky for a time, he became interested in engineering. He received a B.S. degree from the college in 1891 and secured a position there as teacher in elementary mathematics and physics. He continued in that position until 1905, except during 1900–1901, when he served as business agent for the college.

He joined the faculty of the Ohio Mechanics Institute in 1905 as instructor in applied electricity and physics. He was given an M.S. degree by the State College of Kentucky in 1907, and during the scholastic year 1907–1908, on leave of absence from Ohio Mechanics Institute, he took a course in electricity and power plant engineering at the State University of Kentucy which brought him a B.M.E. degree in 1908. He then returned to his duties at Ohio Mechanics Institute, where in 1911 he was appointed dean of the School of Mechanics and Electricity, later becoming head of the Department of Physics and Electricity. He resigned in 1930 after a severe illness. The State University of Kentucky conferred an M.E. degree upon him in 1913.

Professor Muncy became a member of the A.S.M.E. in 1914. He also belonged to the American Institute of Electrical Engineers, the Illuminating Engineering Society, the Royal Society of Arts, American Association for the Advancement of Science, the Engineers Club of Cincinnati, and Tau Beta Pi, honorary engineering fraternity.

Professor Muncy's first wife, Emma (Garred) Muncy, died in 1893, two years after their marriage. He was separated by divorce in 1913 from Mary O. (Hodges), whom he married in 1908.

HENRY HOTCHKISS MURRAY (1871–1934)

Henry Hotchkiss Murray, mechanical engineer and manager for the Mechanical Improvements Corporation, Camden, N. J., died at the Burlington County Hospital, Mount Holly, N. J., on February 28, 1934. Mr. Murray was born at Viola, Kent County, Del., on March 16, 1871, the son of George Henry and Olive I. (Purinton) Murray. He prepared for college at the Hillhouse High School, New Haven, Conn., and was graduated from the Sheffield Scientific School of Yale University with a Ph.B. degree in 1893. The following year he did post-graduate work there in mechanical engineering and was an assistant instructor in shop visiting.

From 1894 to 1906 Mr. Murray was draftsman for McIntosh, Seymour & Co., Auburn, N. Y., with the exception of the first six months of 1896, when he served as acting professor of mechanical and electrical engineering at Delaware College, Newark, Del. From 1906 to 1930 he was connected with the Victor Talking Machine Company, Camden, serving in the capacities of designer, chief draftsman, mechanical engineer, chief engineer, consulting engineer, and manager of the field technical service. He had been with the Mechanical Improvements Corporation since 1930.

Mr. Murray held many patents pertaining to the improvement of the talking machine and a number for camera cases for underwater photography. He was very active in the community of Riverton, N. J., where he made his home. From 1915 to 1923 he was a member of the borough council; from 1916 to 1923, chairman of the Memorial Park Committee; from 1917 to 1934, director of the Cinnaminson Building and Loan Association; from 1926 to 1932, director of the Cinnaminson Bank & Trust Co.; and in 1932–1933, member of the Riverton Board of Education. He served as chairman of the committee on the location of the Philadelphia-Camden bridge of the Camden Chamber of Commerce in 1921, and during the World War was active in Riverton in connection with the sale of liberty bonds and other war work. At that time he was in charge of airplane parts, gun stocks, etc., manufactured by the Victor Talking Machine Company.

Mr. Murray became a junior member of the A.S.M.E. in 1899 and a member in 1905. He also belonged to The Franklin Institute, Yale Engineering Society, Union League of Philadelphia, Society of Colonial Wars, Riverton Country Club, and Yale Club, New York. He married Miss Georgiana Groot in 1898 and is survived by her and their daughter, Cornelia Groot Murray.

ALFRED NATHAN (1866–1933)

Alfred Nathan, president of the Nathan Manufacturing Company, New York, N. Y., died at the Presbyterian Hospital in that city on May 22, 1933, following an operation. He was a native of New York, where he was born on November 21, 1866, the son of Max and Rosalie Nathan.

Following his graduation from Stevens Institute of Technology with an M.E. degree in 1890 he became assistant mechanical engineer for The Geo. F. Blake Manufacturing Co., a position which he held until 1897. He had been associated with the Nathan Manufacturing Company since then, as assistant secretary until 1903, and vice-

president from that time until he became president of the company in 1909.

Mr. Nathan had been a junior member of the A.S.M.E. since 1891. He belonged to the Beta Theta Pi and Sigma Alpha Epsilon fraternities, and to a number of clubs in New York and vicinity, including the Uptown and Criterion clubs, and the Deal and Hollywood golf clubs. He is survived by his widow, Mabel E. (Lauer) Nathan, whom he married in 1892, and by a son, Alfred Nathan, Jr.

HAROLD VAN HOUTEN NEEFUS (1883-1934)

Harold Van Houten Neefus, for many years associated with Francisco and Jacobus, engineers and architects, New York, N. Y., died in Greenwich, Conn., on August 23, 1934, of pneumonia.

Mr. Neefus was born in Newark, N. J., on August 21, 1883, the son of Henry Freeman and Olive (Van Houten) Neefus. After being graduated from the Stevens Institute of Technology with an M.E. degree in 1904, he was employed for three years as a draftsman by the Public Service Corporation of New Jersey. During the next two years he engaged in experimental work, including drafting and shop work, for the Blevney Machine Company, Newark. In 1909 he took a position as assistant to the secretary and general manager of the Foos Gas Engine Company, with which he remained until 1911. He then went to Buffalo to take charge of manufacturing and sales for the Bogart Gas Power Company. In 1914 he became mechanical engineer for the Northern Division, including six plants, of The Texas Company, New York.

During the World War, from 1916 to 1918, Mr. Neefus was engaged in the production of munitions at the Canada Car & Foundry Co. Subsequently he served for two years as special engineer for the City of Chicago in connection with the electrification of the Illinois Central Railroad and the development of the Union Station. He became connected with Francisco and Jacobus in sales and promotion work in 1920 and later was put in charge of the Chicago office and mid-west territory. At the time of his death he was serving as manager for the Eastern District, New Jersey Department, of the Home Owners Loan Corporation, with headquarters at Newark.

Mr. Neefus became a member of the A.S.M.E. in 1915 and belonged to the Delta Tau Delta fraternity. His wife, Grace (Berger) Neefus, died the year following their marriage in 1910. He is survived by a son, Van Houten Neefus.

VICTOR ROBERT NELSON (1898–1934)

Victor Robert Nelson, whose death occurred on October 3, 1934. was born in Cleveland, Ohio, on June 3, 1898, the son of Niels and Annie (Madsen) Nelson. During the summer after his graduation from the Central High School in Cleveland he was employed by the Linde Air Products Company, and he spent subsequent vacations, while attending the Case School of Applied Science, working in various shops for practical experience. He was a member of the Reserve Officers Training Corps at the School in 1917–1918.

In the summer of 1920, followng his graduation with a B.S. degree in mechanical engineering, Mr. Nelson was employed by Van Dorn & Dutton Co., Cleveland, in connection with production routing and time records. In the fall he secured a position in the U.S. Ordnance Engineer's Office in Cleveland, working under the supervision of the resident engineer on the design and specifications for an artillery tractor and other equipment. After he had been there about a year the Cleveland office was discontinued and he was transferred to the Rock Island Arsenal, where he continued in similar work until March, 1922.

He then returned to the Van Dorn & Dutton Co., which in 1928 became a division of Gears & Forgings, Inc. His early work for the company was in the sales and estimating departments. He then became production supervisor and at the time of the merger was put in charge of the organization of a department combining and coordinating the work of the order and production departments, and certain phases of the work of the engineering department. He continued to serve as production engineer for the company until his death, increasing his knowledge in this field by a course in industrial management through the LaSalle Extension University.

Mr. Nelson became an associate-member of the A.S.M.E. in 1931. He had served as a member of the local Committee on Apprenticeship of the National Metal Trades Association, the Committee on Standards of the Cleveland Engineering Society, the Production Standards Committee (as chairman) of the Associated Industries of Cleveland, and the Production Executive Committee of the Cleveland Chamber of Commerce. He belonged to the masonic fraternity.

Mr. Nelson married Miss Ruth Ellen Heininger in 1926 and is survived by her and by two daughters, Margaret Ellen and Karen Elizabeth Nelson.

FRED NOLDE (1874–1934)

Fred Nolde, secretary and treasurer of the Refrigerating Machinery Association, Philadelphia, Pa., died at his home in that city on December 19, 1934. He was born at Norristown, Pa., on January 16, 1874, the son of Henry and Caroline Nolde. He attended the Central High School in Philadelphia and then the Drexel Institute there, from which he was graduated with a B.A. degree in 1891.

Following his graduation Mr. Nolde took a position as draftsman for the Pennsylvania Iron Works Company in Philadelphia. He remained with this company until 1903, being advanced to the position of chief draftsman and sales engineer, and engaging in the design, construction, operation, and sales of cable railroad machinery, steam engines, hydraulic machinery, ice making and refrigerating machinery, and gasoline engines. During the next year he was connected with James Beggs & Co., selling power machinery in New York, and after that held a similar position with the Allis-Chalmers Manufacturing Company, in the Philadelphia district.

From 1905 to 1927 Mr. Nolde was associated with the De La Vergne Machine Company, as sales engineer, chief engineer, and manager in New York until 1926, when he became consulting engineer to the company, with headquarters in Philadelphia. After about two years in that position he became eastern sales agent for the Vilter Manufacturing Company, of Philadelphia, a position which he held until he took the office of secretary and treasurer of the Refrigerating Machinery Association in October, 1927, first on a part-time basis and latterly as a full-time occupation.

Mr. Nolde became a junior member of the A.S.M.E. in 1901. He resigned in 1910 and was reinstated in 1918 as a full member. He was also a member of the American Society of Refrigerating Engineers and belonged to the masonic fraternity and to a number of clubs, including the Engineers' and Bankers in New York. He married Miss Hedwig P. Haenchen in 1913 and is survived by her and their son, Fred Nolde, Jr.

ARVID NORDIN (1881–1933)

Arvid Nordin, whose death occurred on July 19, 1933, had been associated with the M. H. Treadwell Co., New York, N. Y., for nearly twenty years.

Mr. Nordin was born on July 30, 1881, at Westkinde, Sweden, and was educated in that country. After being graduated from the college at Venersborg he studied mechanical engineering for three years at the Polytechnic Institute at Gothenburg. During his college vacations he worked as an apprentice machinist and boilermaker at the Eriksbergs Mechanical Works, Gothenburg, and Nydqvist & Holm Locomotive Works, Trollhattan.

Mr. Nordin came to the United States in 1901 and during the next four years worked as draftsman and designer-checker for the Niles-Bement-Pond Company, Philadelphia, Pa., Westinghouse Electric & Manufacturing Co., Pittsburgh, Pa., Carnegie Steel Company, Sharon, Pa., Allis-Chalmers Company, Milwaukee, Wis., Illinois Steel Company, South Chicago, and Sargent & Lundy, Chicago, Ill. From 1905 to 1912 he was connected with the Tennessee Copper Company, Copperhill, Tenn., in the positions of designer-checker, chief draftsman, and mechanical engineer. From then until he took up his work with the Treadwell organization he was engineer for the Link-Belt Company, Chicago.

During his association with the M. H. Treadwell Company, Mr. Nordin devoted a large measure of his ability and energy to the development of the various items of hydraulic equipment which they manufacture. He was the inventor of many improvements in the design and details of spillway and intake gates, hoists and accessory equipment, and held many valuable patents on these improvements. As contracting engineer for the company, he enjoyed an unusually wide acquaintance among hydraulic engineers and in the hydroelectric power industry in general. In recognition of his ability and accumulated experience with respect to hydraulic equipment, his advice and opinions were in constant demand.

Mr. Nordin became a member of the A.S.M.E. in 1913. He is survived by his widow, Ruth L. Nordin, of Allegan, Mich., and by a son, John Arvid Nordin.

EDWARD WILLIAM O'BRIEN (1883–1934)

Edward William O'Brien, who died at St. Joseph's Hospital in Providence, R. I., on August 4, 1934, was born in that city on November 29, 1883. His parents were Edward and Mary (Lindsay) O'Brien. Mr. O'Brien attended the Technical High School in Providence and was graduated from the evening course at the Rhode Island School of Design in 1903. Later he took special courses at Franklin Union, Boston, and the Boston Y.M.C.A. in applied mechanics, hydraulics, and other subjects.

From 1903 to 1907 Mr. O'Brien was employed by Knight C. Richmond, engineer and architect of Providence, as a draftsman and designer, working on power plants and industrial buildings. He spent the next two years in Fitchburg, Mass., at the D. M. Dillon Steam Boiler Works, engaged in special work for the New England Boiler Manufacturers Association on boiler design standardization, and compiling a catalog for the company.

During the years 1910–1917, with the exception of about a year, he was associated with Charles H. Fish, consulting engineer of Boston, working part of the time in the office on the design of industrial buildings, and making appraisals and reports, and part of the time in the field, supervising construction. In 1913–1914 he was engaged by the General Electric Company at Lynn to design and serve as inspector during the construction of a reinforced-concrete storehouse.

The Harry M. Hope Engineering Co., Boston, employed Mr. O'Brien from 1919 to 1921, keeping him in the field the greater part of the time. He spent about eight months as general inspector during the construction of a large power plant in Connecticut, and more than a year in St. Louis, Mo., in connection with the construction and equipment of a power house and three eight-story reinforced-concrete buildings for the United Drug Company.

He was next connected with the American Printing Company, Fall River, Mass., as assistant to the plant engineer. His most important work for this company, with which he remained until 1930, was the supervision of the design, construction, and equipment of a large cotton mill in Tennessee, including the mill itself, and a storehouse, power plant, and pumping station, as well as some two hundred dwellings for mill operatives, with water and sewer systems.

Since 1930 Mr. O'Brien had been connected with the Public Buildings Department of the City of Providence, being second deputy to the inspector of public buildings, in charge of zoning, at the time of his death.

Mr. O'Brien became a member of the A.S.M.E. in 1927. He is survived by his widow, Gertrude (Curran) O'Brien, whom he married in 1926, and by two children, Margaret J. and Robert E. O'Brien.

SPENCER OTIS (1858–1933)

Spencer Otis, whose death occurred on June 10, 1933, was born in Orange, N. J., on February 26, 1858, the son of Daniel Carmichael and Clarissa Otis. He entered Amherst College in 1876 but was obliged to leave because of illness. The following year he secured work at a brick and terra cotta factory in Louisville, Ky., and in 1877 he began an apprenticeship with the Union Pacific Railroad at its Omaha, Neb., shops. Upon the completion of his training he worked for that road for some years, advancing to the position of master mechanic.

In 1888 he became general manager of the Kansas City Switch & Frog Co., and two years later president of the Phoenix Foundry Company. He engaged in consulting work, beginning in 1891, and five years later organized the Spencer Otis Company, of Chicago, manufacturers and agents for railway supplies. He served as president of this company until its sale some twelve or fifteen years later. For a number of years, beginning in 1892, he was also president of the National Dump Car Company, Chicago, which in 1904 was combined with the Rodger Ballast Car Company, with Mr. Otis as vice-president. He held many patents pertaining to railway cars and equipment.

About 1902 Mr. Otis became interested in large-scale farming, and much of the latter part of his life was devoted to it, especially to the development of pastures in the Middle West. His chief engineering connection in recent years was with the National Boiler Washing Company, of which he became president in 1919, and the Locomotive Terminal Improvement Company, which he formed in 1925 to manufacture plants for the former company, and to develop and build plants for locomotive terminals. At the time of his death he had retired and was residing in Waco, Texas, with a married daughter.

Mr. Otis became a member of the A.S.M.E. in 1890 and also belonged to the Western Railway Club, the Union League Club, Chicago, and Barrington Hills Country Club.

He was twice married, his first wife, Eleanor G. Beard, dying in 1885 and his second, Julia Melchers, in 1902. He is survived by one son, Spencer Otis, Jr., of Chicago, and by five daughters.

GEORGE BAILEY PAGE (1887–1934)

George Bailey Page, Philadelphia district manager for The Terry Steam Turbine Company, of Hartford, Conn., died at his office from a hemorrhage on September 10, 1934. His widow, Helen (Ferguson) Page, whom he married in 1919, and two children, George F. and Jane Helen Page, survive him.

Mr. Page was born at Williamsport, Pa., on April 27, 1887, the son of Earnest A. and Sarah (Bailey) Page. He attended the Dickinson Seminary in Williamsport and was graduated from Cornell University with a mechanical engineering degree in 1911. He then served an apprenticeship at the DeLaval Steam Turbine Company, Trenton, N. J., and worked for the company in different capacities until early in 1916. During the next five years he was district engineer for the Rush Machinery Company, Pittsburgh, Pa., holding the honorary title of vice-president. In 1921-1922 he was Pittsburgh district sales engineer for the Underfeed Stoker Company of America and then held a similar position with the Sanford Riley Stoker Company for nearly two years.

Between then and 1926 Mr. Page was a member of the firm of Geo. B. Page & Co., in Pittsburgh, leaving there to become associated with B. F. Sturtevant Co., New York. He had been with The Terry Steam Turbine Company since January 1, 1932. He was well versed in both sales and installation work in his field.

Mr. Page became an associate-member of the A.S.M.E. in 1924, and had belonged to the Engineering Society of Western Pennsylvania and the Association of Iron and Steel Electrical Engineers, and to Alpha Chi Rho fraternity.

GEORGE EDWARD PERKINS (1877–1934)

George Edward Perkins, who died at the Brooklyn Hospital on May 4, 1934, was born in Newburyport, Mass., on November 23, 1877, the son of George Collyer and Sarah (Hinckley) Perkins. He attended high school in Amesbury, Mass., for three years, and then found work in the boiler and dynamo rooms of the electric light company there. From June, 1900, to November, 1902, he secured varied experience as operating engineer and electrician in resort hotels in New England and Florida. He then became operating engineer in charge of the steam plant of W. W. Cross & Co., Inc., Brockton, Mass., with which he remained until 1909, and subsequently held a similar position with the T. D. Barry Co., also in Brockton.

Mr. Perkins entered the employ of the Fidelity & Casualty Co. of New York, in November, 1911, as a boiler inspector. About a year later he was transferred to the home office of the company as assistant to the chief boiler inspector. He remained with the company until the end of January, 1920, his work in connection with the inspectors and the insured requiring a knowledge of the design, construction, and operation of steam boilers and engines. Since then he had been assistant engineer, in the Engineering Department, of the Royal Indemnity Company, New York, his duties being similar to those of his previous position, but involving more responsibility.

Mr. Perkins became a member of the A.S.M.E. in 1925, and was active in the masonic fraternity and Odd Fellows, holding several offices. He is survived by his widow, Alice L. (Duston) Perkins, whom he married in 1904, and by a daughter, Helen M. Perkins.

GEORGE SIMEON PERKINS (1868–1933)

George Simeon Perkins, who had been engaged in contracting and superintending construction in and about San Francisco, Calif., for some years, died suddenly on June 21, 1933, at his home in Oakland, Calif. He had been in failing health for two years.

Mr. Perkins was born at Bridgeport, Conn., on August 14, 1868, the son of George B. and Maryetta (Blood) Perkins. After completing his high school course in that city he attended the Stevens Institute of Technology, Hoboken, N. J., securing a degree in mechanical engineering in 1891. During the next six years he worked for the Dow Typesetting & Composing Co., New York, the Philadelphia Traction Company, and the East Jersey Water Company, in connection with the installation of pumping engines at Little Falls, N. J. Subsequently he was employed by the Ingersoll-Sargeant Company, on central station air plants, and by several portland cement plants no longer in existence, including the Pacific, Hecla, and Midland.

Mr. Perkins became a member of the A.S.M.E. in 1926. He is survived by his widow, Adela B. Gayer, and by three children, Daniel Carter, George Stephen, and Mary Gayer Perkins.

JOHN McCLARY PERKINS (1878–1934)

John McClary Perkins, treasurer of the Wittenmeier Machinery Co. of N. Y., New York, N. Y., died in the New York Hospital on August 10, 1934, of pneumonia.

Mr. Perkins was born in Washington, D. C., on September 4, 1878, the son of John McClary and Lucy (Flagg) Perkins. He attended high school at Arlington, Mass., and was graduated from the Massachusetts Institute of Technology in 1901. During the next three years he worked in the steel foundries of the Everett, Mass., branch of the General Electric Company, becoming a foreman molder. He

then took a position as foundry superintendent at the Saco-Lowell Shops, where he installed a complete foundry accounting and cost system and put the foundry on a piece-work basis. He left there in 1909 to build, equip, and put into operation a foundry for the Chalmers Motor Company.

From 1911 to 1916 he was connected with Turner & Seymour, Torrington, Conn., first as foundry manager, and for the last three years as sales manager. He then became works manager of the Gilbert & Barker Manufacturing Co., a subsidiary of the Standard Oil Company, and put into successful operation there the conference committee plan later adopted by all branches of the Standard Oil. In 1919 he went to the Kewanee Works of the Walworth Manufacturing Company, where he settled a strike and formed a conference committee. He became manager of the McNab & Harlin Manufacturing Co., Paterson, N. J., in 1921, and remained there until about 1923, when he became treasurer of the Lapham-Perkins Engineering Company, of New York. He had been treasurer of the Wittenmeier Machinery Company since 1926.

Mr. Perkins became a member of the A.S.M.E. in 1923, and was a former president of the Refrigeration Association of New York. He also belonged to the masonic fraternity, Elks, and New York Rotary Club. He is survived by his widow, Floye (Dinwiddie) Perkins, whom he married in 1906, and by a son, Wilkes D. Perkins.

ARTHUR FRENCH POOLE (1872–1934)

Arthur French Poole, who had been granted many patents for his inventions in connection with electric clocks and adding machines, died at his home in Ithaca, N. Y., on April 26, 1934, after a short illness.

Mr. Poole was born in Cumberland, Md., on August 21, 1872, son of Arthur A. and Virginia (French) Poole. He received his early education there. Later the family moved to Washington, Pa., where he learned the watchmaker's trade under his father, then entered Washington and Jefferson College, with the class of 1891, and studied there for three years. Having become interested in the newly founded Leland Stanford Jr. University, he went there in 1894, taking an A.B. degree in astronomy in 1896. At Stanford he made use of his watchmaking training to work his way through college. He also spent considerable time at Lick Observatory.

Mr. Poole returned to Pennsylvania after his graduation and went to work as a draftsman for John Brashear, noted maker of telescope lenses and scientific instruments, at Alleghany. He left in 1897 to work on a clock system of his own invention and in 1899 formed the Poole Clock Company, Wheeling, W. Va., to manufacture the clock. The company was dissolved in 1902, because of business conditions. His interest turning to electrical work, he joined the Mountain State Electric Company, of Wheeling, to do development work in connection with automatic telephones, and party line systems. Many of his early patents reflect his work here.

In 1908 Mr. Poole went to Chicago, Ill., where for several years he engaged in special development and research work. He worked for Jackson and Jackson on a workmen's time register, for the Holtzer Cabot Company, of Boston, on a frequency meter, and independently on a speedometer. In 1911 he took charge of the patent and development department of the Wahl Adding Machine Company there, and some of his most brilliant work was done in the development of adding machine attachments for typewriters. In 1918 the company acquired patents on automatic pencils, which enabled it to develop and market the Eversharp pencil, and Mr. Poole's inventive genius assisted materially in this work. He was made a director and vice-president of the company in 1920, and two years later, when the Remington Accounting Machine Corporation, a subsidiary of the Remington Typewriter Company, was formed to take over the Wahl adding machine business, he went to New York as vice-president and general manager of the new organization. He remained active as a director of the Wahl company, however, until his death.

Perhaps due to his early training, he was always deeply interested in clocks, especially electric clocks and electric clock systems. His first patent, granted in 1899, was for an electric clock. He was a pioneer in the development of the synchronous motor clock, and methods of frequency control for them, and many of the earliest patents in this field were granted to him. In 1924 he formed the Poole Manufacturing Company for the purpose of making a battery clock of his invention. He established a factory at Westport, Conn., and manufactured these clocks there until 1926, when the business was affiliated with the Morse Chain Company, of Ithaca, N. Y., and moved to Ithaca. His work in inventing and designing this clock led The Franklin Institute to award him a Certificate of Merit in 1933.

Mr. Poole retired from active business in 1931, but continued in development work. At the time of his death he had many patents pending in the patent office. Since his first patent was granted in 1899, he had received more than one hundred, chiefly in the adding machine and electric clock fields.

Like many scientists, Mr. Poole was a lover of music. He had a large collection of symphonic scores, played the 'cello and piano, and while in Chicago was a member of a string quartet.

Mr. Poole became a member of the A.S.M.E. in 1921 and also belonged to The Franklin Institute.

Surviving Mr. Poole are his widow, Maryline (Barnard) Poole, whom he married in 1904; two daughters, Mrs. Martin D. Hardin, Jr., of New York, N. Y., and Miss Elizabeth Poole, of Ithaca; and three sons, Frank and John Poole of Ithaca, and Arthur B. Poole, of Bristol, Conn.

WILLIAM BANCROFT POTTER (1863–1934)

William Bancroft Potter, for many years engineer for the railway department of the General Electric Company, Schenectady, N. Y., died at his home in that city on January 15, 1934. Mr. Potter was one of the best known electric traction engineers in the United States. His career went back prior to 1890, when electric street cars were still struggling to overcome technical difficulties and public prejudice. His contributions to the electric traction art were substantial and long continuing. Many of them were covered by the 130 patents which had been issued in his name.

Mr. Potter was born on February 19, 1863, at Thomaston, Conn., the son of Horace and Charlotte (Pierce) Potter. He gave early evidence of his mechanical aptitude and during the latter years of his school life spent his vacations working in the Seth Thomas clock factory. After completing the Thomaston high-school course he served an apprenticeship as a machinist with Sawtell and Judd, in Hartford, Conn. He continued with that company for two years after the completion of his apprenticeship, securing further training in machine shop and steam engineering work. While repairing stationary engines at the plant of the Hartford Electric Light Company he became interested in electrical apparatus. He foresaw that the development of the electric dynamo would have a marked influence upon the future of the steam engine, and this trend in his thinking led him to study dynamos and electric generating stations. In the summer of 1887 he entered the employ of the Thomson-Houston Electric Company, Lynn, Mass. After a period of shop training he was sent into the field to adjust trouble in electric light plants which the company had installed, and served for a time as superintendent of the electric light company at Raleigh, N. C.

In the fall of 1889, having seen his first electric street car, he returned to Lynn and became a shop student in electric traction work. This was again supplemented by practical experience in the field, particularly on the West End Street Railway Company's system in Boston, and others of less importance.

Later, returning to the Lynn plant, he aided in the development of the Thomson-Houston single-reduction motor for street cars, and when Walter H. Knight was appointed engineer of the railway department he became Mr. Knight's principal assistant. He was active in working out railway circuit breakers, fuse blocks, lightning arresters, and switches. The most serious problem in electric railway work at that period was the matter of an efficient controller for electric cars. Knight and Potter both worked on this, and ultimately they solved the problem by the invention of the series-parallel controller, which became the basic method in use all over the country from that time forward. Mr. Potter's part in the development of the controller led to his being placed in charge of all control work for street railways.

In 1894, when the Thomson-Houston Electric Company became part of the General Electric Company, Mr. Potter was transferred to Schenectady, and in 1895 he was appointed engineer of the railway department, succeeding Mr. Knight. He continued in this position until his retirement in 1930. In that long interval he witnessed and aided materially the tremendous expansion of electric traction in the street railway and interurban field, and also participated in the earliest installations of electrical equipment upon the lines of steam railroads.

The major projects in which he was associated were the electrification of the Manhattan Elevated Railway of New York, the electrification of the Grand Central Terminal, and the Paris and Orleans railroad in France. Other projects with which he was prominently identified included the West Jersey and Seashore, Detroit tunnel, Great Northern Railroad, Southern Pacific, Butte, Anaconda and Pacific and Milwaukee, Chicago, St. Paul and Pacific railroads.

A suggestion made by Mr. Potter in the spring of 1896 resulted in the establishment of the old General Electric test railroad, or test tracks, along the berm bank of the Erie canal, stretching outward from the local General Electric plant about a mile and a half. There all tests of electric railway equipment were conducted until 1919, when the berm bank track was discontinued.

On this track were tested early types of gas-electric rail cars, some of which later went into commercial service on eastern railroad lines. Mr. Potter participated in many of these test trips and had an active part in directing the development of the propulsion equipment for these cars.

Possibly his most valuable work within the General Electric Company during latter years had been his organization and development of a great engineering staff for electric railway and electric railroad work. His supervision of the department of which he was the head was practically an unbroken stretch of harmonious relations.

Mr. Potter became a member of the A.S.M.E. in 1897. He also belonged to the American Society of Civil Engineers, American Institute of Electrical Engineers, and Society of Naval Architects and Marine Engineers. He served as a member of the A.I.E.E. traction and transportation committee during 1916–1917, 1919–1921, and 1923–1924; its transportation committee, 1925–1932; and its education committee the year before his death. He had contributed many articles to the technical societies.

Mr. Potter had been interested for many years in community life and the development of Schenectady. He particularly followed the fortunes of the Schenectady Civic Players. He was a man of various hobbies, which gave him a means on many occasions of delighting his friends. He was a consistent radio enthusiast from the early days of the art, and had done independent research and development work on a type of radio receiver.

Mr. Potter's club memberships included the Engineers', Transportation, and Railroad, New York; Mohawk, Mohawk Golf, and Edison, Schenectady; and Griswold, Erie. He was twice married, his first wife being Loretta Harward, of Raleigh, N. C., whom he married in 1890, and his second, who survives him, Rose Hubbard, formerly of Sandusky, Ohio, whom he married in 1912. A daughter, Dorothy (Potter) Bird, also survives him.

THOMAS PROSSER (1854–1934)

Thomas Prosser, senior member of the firm of Thomas Prosser & Son, metal dealers, New York, N. Y., died on December 23, 1934, at his home in Garden City, L. I., N. Y., after a long illness. A daughter, Mrs. John Good, and a son, Thomas H. Prosser, both of Garden City, survive him, as do three sisters and four brothers.

Mr. Prosser, who was the eldest of eleven children, was born in Brooklyn, N. Y., on August 24, 1854. He attended the Lafayette and Brooklyn Polytechnic Institutes and at the age of twenty-one became a member of the firm, which was established by his grandfather. He made frequent trips abroad in connection with business for the Krupp Steel Works, of Essen, Germany, for which the firm became American representative in 1848, and also traveled extensively in the United States.

Mr. Prosser became a member of the A.S.M.E. in 1893 and also belonged to the Railroad and Engineers Clubs and the Chamber of Commerce of New York.

FREDERICK GEORGE PROUTT (1870–1934)

Frederick George Proutt, consulting engineering and chairman of the City Board of Water Commissioners of Memphis, Tenn., died in that city on May 27, 1934, after an illness of six months. Mr. Proutt was born at Bowmanville, Ont., Canada, on November 27, 1870, the son of Mark James and Martha (Burke) Proutt. His mother died at his birth. He secured his early education in Canadian public schools and at the age of seventeen entered the employ of the Canadian Bell Telephone Company. He worked there until 1890, when he took a position as superintendent of the Bowmanville Electric Light Company.

In 1892 Mr. Proutt came to the United States and for two years studied electrical engineering at the works of the General Electric Company at Lynn, Mass. Upon completion of the course in 1894 he became chief electrician and assistant to the general manager of the Malden Electric Company, Malden, Mass. He left New England in 1897, accepting the position of chief electrician of the Memphis Consolidated Gas & Electric Co., of which he later became general superintendent. During the nine years he spent with this company he designed plans for and supervised the installation of the first extensive underground conduit system for distribution of electricity in the city.

In 1906–1907 Mr. Proutt was general manager for the Electric Railway Light & Power Co. at Jackson, Miss., for which he designed and installed electric power and gas plants. He returned to Memphis at the beginning of 1908 and opened an office for consulting engineering practice, specializing in the design and management of public utilities. He designed a number of plants in Arkansas, Mississippi, and Tennessee, including light and heat plants for the University of Mississippi, Light and Power for the Brownsville Cotton Oil & Ice Co., water works and sewage disposal systems for the city of Brownsville, and other public utility plants at Friar Point, Laurel, and Cleveland, Miss., and at Blytheville, Osceola, and Luxora, Ark., as well as private installations in Memphis and vicinity.

In May, 1917, during the World War, Mr. Proutt entered the First Officers Training Camp at Chickamauga Park, Ga. He was commissioned a captain of engineers but subsequently discharged because of defective eyesight. He was then appointed consulting engineer for the War Finance Corporation at Washington, D. C., and devoted two years to Government work in this civil capacity, part of the time being spent in Washington and the remainder in Memphis. His work included a survey of Tennessee industries for the Naval Consulting Board.

Mr. Proutt became chairman of the Memphis City Board of Water Commissioners in 1920, and held that office until his death. His able representation of the city in 1920 in the appraisal of the plants of the Memphis Consolidated Gas & Electric Co., which were subsequently acquired by the Memphis Power & Light Co., led also to his selection in 1921 as consulting engineer for the State Railroad and Public Utilities Commission of Tennessee; this connection also continued throughout the remainder of his life.

Mr. Proutt had also served as president of the Consumers Water Company, at Blytheville, and as treasurer of the Home Light & Ice Co., at Cleveland. At the time of his death he was president of the Peoples Finance & Thrift Co., at Memphis, and a trustee of the William R. Moore School of Technology, and for five years he was a trustee of the Memphis and Shelby County Tubercular Hospital.

In 1914 Mr Proutt aided in organizing and was the first president of the Engineers' Club of Memphis. To all club activities he was generous with his time and resources, the Technical Education Fund, which since his death has been given his name, being his chief interest. He served as chairman of this fund from the time of its inception.

He was also a charter member of the Memphis Section of the American Institute of Electrical Engineers. He became a member of the A.S.M.E. in 1915. He was a vestryman at St. Luke's Episcopal Church in Memphis and had attained the highest degrees in Masonry.

Surviving Mr. Proutt are his widow, the former Miss Laura J. Yarnold, of Whitby, Ontario, Canada, whom he married in 1895, and a daughter, Miss Jean Proutt. A younger daughter, Marjorie, died in 1925.

MARK ANTHONY REPLOGLE (1861–1934)

Mark Anthony Replogle, president of the Replogle Heater Company, Akron, Ohio, died at his home in that city on May 25, 1934, after a year of poor health. He was born at Martinsburg, Blair County, Pa., on September 8, 1861, son of Samuel B. and Margaret A. (Hanawalt) Replogle. His father was the inventor of an automatic buffer and coupling widely used on railroads. His childhood was spent in Mifflin County, Pennsylvania, where he attended the public schools. Subsequently he taught in the schools there for a time, then went to Iowa, where, after a period on a farm, he worked in machine shops in Hampton and Ackley.

In 1885 Mr. Replogle entered the Iowa State Normal School at Cedar Falls, where he took a three-year mechanical course, spending his vacations as representative of the Akron, Ohio, firm of Aultman Miller & Co. in the harvest fields around Cedar Falls. During the next two years he again taught school in Pennsylvania, then became associated with H. E. Olbrich and H. H. Clay, of Cedar Falls, in the manufacture and sale of water-wheel governors. In 1895 the manufacturing rights were purchased by the Selle Gear Company, of Akron, and the Replogle Governor Works was organized, with Mr. Replogle as chief engineer. The Selle Gear Company was later absorbed by the Webster, Camp & Lane Co., for which he became hydraulic engineer in 1899. In 1906 he organized the Lombard-Replogle Engineering Company, specializing in water-wheel governors. The Replogle Heater Company was formed in 1925.

Mr. Replogle's interest in water wheels went back to his childhood, when he made wheels which he tried out in neighborhood streams. From 1889 on he devoted himself to design in this and related fields. He held patents in the United States and other countries for his original inventions, which include a discharge gate for turbine wheels, an electrical governor, a mechanical speed governor, a sliding toggle, mechanical movements, and a smokeless home heater. He was associated with the design and construction of a number of water-power plants, including one at Sault Ste. Marie, Mich., while he was with the Webster, Camp & Lane Co., which was the largest erected in the United States up to that time. He furnished the governors and turned on the water that started the first turbines of the Niagara Falls Paper Company.

In 1894 he published a treatise entitled, "Electricity and Water-Power and Their Inter-relations," republished by the Electrical Re-

view Publishing Company, New York, in 1896, and his paper on "Speed Regulation in Water-Power Plants," prepared at the request of The Franklin Institute, was one of the earliest works in this country on that subject. Other papers by him have appeared from time to time in the technical press in this and other countries.

Mr. Replogle became a member of the A.S.M.E. in 1898. He was a 32d degree Mason. He is survived by a son, George B. Replogle, and by two daughters, Mrs. Edith (Replogle) Harley and Mrs. Wilhelmina (Replogle) Groff, all by his first wife, Emma H. (Steeley) Replogle. His second wife, Carrie (Pardee) Replogle, died in December, 1933.

MYRON B. REYNOLDS (1880–1934)

Myron B. Reynolds, city engineer of Chicago, Ill., died at the Edward Hines Hospital in that city on January 27, 1934, of cancer of the throat. He had been in the service of the city, in various capacities, since 1907, although inactive because of his illness for some time prior to his death.

Mr. Reynolds was born at Pana, Ill., on December 12, 1880. He entered college at Rose Polytechnic Institute in 1900 and spent two years there. He then worked for a year as draftsman on elevated railway rolling stock and structures in Chicago and a year in St. Paul, part of the time on bridge design for the Great Northern Railway and the remainder as draftsman in the Public Works Department of St. Paul. He reentered college as a junior at Armour Institute in 1904 and was graduated two years later.

Immediately following his graduation Mr. Reynolds entered the employ of the City of Chicago. During the next three years he worked as engineer in the Bureau of Engineering, in connection with tunnel construction, and as office assistant to the assistant city engineer. He then spent two years as superintendent of concrete bridge construction for a contracting engineer in Chicago. He returned to the employ of the city in 1911 in the position of assistant engineer of the Bureau of Engineering. After two years he was made engineer of water-works design and held that position until 1923, when he became assistant city engineer. In 1917 he was granted leave of absence for military duty and served as a major in the Corps of Engineers for about fifteen months, being stationed at several different camps in the United States.

In 1927, after the death of John Ericson, under whom he had been serving as assistant city engineer, Mr. Reynolds was called upon to fill the position of acting city engineer for several months and in 1931 he was appointed city engineer. Many of his recommendations have been adopted in modernizing the Chicago water-works system, particularly in connection with metering and filtration projects. He was senior author, with Arthur E. Gorman, city filtration engineer, of a paper on "Chicago's Water Pollution Problem, Past, Present, and Future," which brought him the award, just a few days before his death, of the medal of the Illinois Society of Engineers for the best paper published by one of its members in 1933.

Mr. Reynolds became a member of the A.S.M.E. in 1924. He also belonged to the American Water Works Association and was commander of a post of professional engineers of the American Legion in Chicago. He is survived by his widow, Irene Reynolds, and by a son, Myron, Jr.

PALMER CHAMBERLAINE RICKETTS (1856–1934)

Palmer Chamberlaine Ricketts, president and director of the Rensselaer Polytechnic Institute, which he had served as instructor to president during sixty successive years, and honorary member of the A.S.M.E., died at the Johns Hopkins Hospital, Baltimore, Md., on December 10, 1934.

From an institution of less than two hundred students he had seen Rensselaer grow to seventeen hundred students and from two buildings to twenty or more. For most of this growth he was largely responsible, owing to his scholarship, foresight, and administrative ability. Of the present buildings used by the Institute, only three were erected before his presidency. At present there are twelve distinct courses offered by the Institute, of which only one was offered at the time he undertook the direction of the Institute. This expansion has been the result of his conception of the field of endeavor of a true polytechnic institute.

Palmer Ricketts, the son of Palmer C. and Eliza (Getty) Ricketts, was born at Elkton, Cecil County, Maryland, on January 17, 1856. His early boyhood was spent in Maryland, but in 1865 after the death of his father the family moved to Princeton, N. J., and here he was prepared for college by Mr. John Howard O'Brien, a tutor in Latin of Princeton College, and entered the Rensselaer Polytechnic Institute in September, 1871, at the early age of fifteen.

His high scholastic record was remarkable and was gained while he participated in the student activities of that day. In 1875, at the age of nineteen, he received his degree of Civil Engineer with twenty-three other young men.

In 1874, during the student days of Palmer Ricketts, the Institute celebrated the Fiftieth Anniversary of its founding and from the student body of less than 200 members, eleven men received the degree of Civil Engineer in that year. The C.E. degree was the only degree conferred by the Institute at that time.

On account of his high scholarship Palmer Ricketts was offered an instructorship in mathematics and astronomy, which so appealed to him that he accepted this rather than entering at once into the practice of engineering in the activities of that day, although he carried on practical work in connection with his teaching, as assistant engineer of the Troy and Boston Railroad in 1876.

In 1882 he was made an assistant professor of mathematics and astronomy, and in 1884 he was appointed the William Howard Hart Professor of Rational and Technical Mechanics, which had previously been held by the late William H. Burr. The success of his teaching in structures and hydraulics was so outstanding that he was a consultant in bridge engineering of the Rome, Watertown, and Ogdensburg Railroad from 1887 to 1891, the engineer of the Public Improvement Commission of Troy from 1891 to 1892, the River Commission of Corning from 1897 to 1898, and the designer and supervisor of erection of a number of bridges and hydraulic structures.

His work at the Institute prepared him as an expert in patent cases and he presented testimony in important litigations between the years 1886 to 1897.

In 1892, after the resignation of Director D. M. Green, Palmer Ricketts, at the age of thirty-six, became the director of the Institute under the presidency of John Hudson Peck, a lawyer of the City of Troy, N. Y. The separation of the directorship from the presidency caused much difficulty in the management and development of the Institute, owing to the lack of understanding of the educational needs of the Institute by the presiding officer of the Board, so that the early days of Director Ricketts were filled with trips from the "Hill" to the city with requests and arguments. Fortunately, in 1901 the two offices were combined and Director Ricketts became president and director. Although president for the last thirty years he was always affectionately called "the Director," a title which his loyal friends used, and, we believe, he liked.

On November 12, 1902, Director Ricketts married Miss Vjera Conine Renshaw, of Baltimore, Md., and this wedding was marked by demonstrations of affection and admiration for the Director by the faculty and student body. This affection and admiration for him has been extended by the faculty and students to his wife, who entered into and shared all of his plans and activities during the full term of his presidency.

In 1904 the burning of the main building was the beginning of a new Institute, the monument to the ability, wisdom, and perseverance of Director Ricketts. Such a catastrophe as the fire is often a blessing in disguise. It proved so in this case. With the major part of the plant destroyed, an offer was made to transfer the Institute to a large university with the retention of the name of the Rensselaer Polytechnic Institute, but President Ricketts with his faculty determined to remain at Troy and build again on the foundations established three-quarters of a century before.

After a conference with the late Capt. Robert W. Hunt, Director Ricketts went to Andrew Carnegie and persuaded him to give $125,000 for a new building. Then money was raised to purchase the Warren House and grounds on Eighth Street and money was given by Mr. J. J. Albright for part of the cost of the William Weightman Walker Laboratory, named in memory of a graduate of the class of 1886.

Director Ricketts then persuaded Mrs. Russell Sage to give a sum of one million dollars to found a course in mechanical engineering as a memorial to her husband. This was in the year 1907 and at this time the Institute, under the guidance of the Director, inaugurated the courses in mechanical engineering and electrical engineering.

From here on the years have brought to the Institute gifts for buildings and endowment from alumni, friends, and citizens of Troy and an enlargement of its field of education. A recital of this progress would show the regard and trust which these donors have had in the man who indicated the needs of the institution which he had loved, served, and directed for more than a generation.

Following the introduction of courses in mechanical and in electrical engineering, the course in chemical engineering, and, subsequent to 1922, courses, in arts, science, and business administration, in physics, in chemistry, in biology, in architecture, in metallurgical engineering, in industrial engineering, and in aeronautical engineering have been added. A new building which will house the aeronautical, industrial, and metallurgical departments, will be called the Ricketts Building, in his honor.

In all of this development Director Ricketts was guided by the

thought of service. His theory of admissions and his desire to increase the work offered by the Institute was inspired by his desire to give to those who wanted an education in some field of technology a chance to prove their worth. He always thought that offering a chance to those who might make good was more important than the insistence upon formal admission examinations.

In the administration of the Institute, he was ever the champion of those who did not receive the "fair deal." Although believing in student activities and student life, he was not willing to have hazing or freshman rules interfere with the free life of every member of the student body. His stand against hazing came only when this form of student life went beyond bounds of decency, fair play, and even morality. He was ever willing to remain firm to his convictions, even under adverse criticism from students or alumni. His ability to handle student discipline was remarkable and effective. In its administration he had his fixed conviction, but his great sense of humor was a wonderful aid in this work.

He was interested in the possibility of aid, financial and scholastic, for the students needing such. To him personally many owe their graduation, for without his aid it would have been impossible to continue.

Director Ricketts stood for the highest scholarship from students and faculty, and never did he deviate from the motto of the Institute, "Knowledge and Thoroughness," in his administration of Institute educational policies. He insisted on the traditions of daily recitations, and topic work and even in the "scholastic amusements," as Amos Eaton called the work in the laboratories, the work had to be tested by final examination.

In 1924 the Institute celebrated the Centennial of its founding and Director Ricketts was most happy in the recognition which came to the Institute from governments, prominent engineers, and educational institutions of the whole world through the greetings of those in attendance at these extensive ceremonies.

Director Ricketts was active in civic affairs as shown by the positions held at the time of his death. He was a trustee and vice-president of the Troy Public Library, a trustee of Dudley Observatory of Albany, N. Y., of the Albany Academy, of the Albany Medical College, and the New York State College for Teachers. He was a director of the Samaritan Hospital of Troy and vice-president of the Rensselaer County Tuberculosis and Public Health Association. He was also a Director of the National City Bank of Troy.

In 1905 Stevens Institute of Technology conferred upon Director Ricketts the degree of Doctor of Engineering and in six years later New York University conferred the degree of Doctor of Laws upon him.

Director Ricketts was a member of the American Society of Civil Engineers for many years, a director in 1899–1901, and a vice-president in 1916–1917. His connection with The American Society of Mechanical Engineers extended over forty-four years, and in 1932 he was elected to Honorary Membership. Following this election by our Society he was also elected an honorary member of the American Society of Civil Engineers. He was a commander of the Order of the Crown of Italy and a Commander of the Legion of Honor (France). He was a fellow of the American Association for the Advancement of Science and a member of the American Philosophical Society. He was also a member of the American Institute of Mining and Metallurgical Engineers, the Institution of Civil Engineers of Great Britain, and Sigma Xi.

Director Ricketts has written two histories of the Rensselaer Polytechnic Institute and a number of papers have been contributed by him to the State Commission on Railroads and on Education, to the technical and daily press, and to the Proceedings of technical societies. In all of these he was a clear and forceful writer, free from all sham and uncertain expressions.

Not only was Director Ricketts a man of great foresight, administrative ability, and perseverance but he was a most helpful leader under whom to work. He would listen carefully to your plans and suggestions and then give you the authority to carry out plans which he approved. He never withheld praise for good work and in many cases where adverse criticism might have been offered by others he would take the responsibility himself and assume any blame attached to the transaction. His faculty and his trustees were always ready to follow his lead after a decision had been reached by discussion. His force of character, his human interests, and his devotion to Rensselaer remain as a cherished memory and guiding influence for our work in the future.

As was written of Wren in St. Paul's so of Ricketts at Rensselaer we may write in clear letters: "If you ask for my monument look about you."—Arthur M. Greene, Jr.[1]

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[1] Dean, School of Engineering, Princeton University, Princeton, N. J. Past Vice-President, A.S.M.E.

WILLIAM ROBINSON (1876–1934)

William Robinson, master mechanic for the Eberhard Faber Pencil Company, Brooklyn, N. Y., died in that city on December 31, 1934, after an illness of several months. He is survived by his widow, Elizabeth (Thomson) Robinson, whom he married in 1906, and by two daughters, Mrs. Helen (Robinson) Fairchild and Miss Dorothy T. Robinson.

Mr. Robinson was born in Calderbank, Scotland, on March 3, 1876, son of Andrew and Margaret (Watson) Robinson. He was educated in the United States, attending grammar school and evening high school in Brooklyn, and taking a mechanical engineering course through the Alexander Hamilton Institute. He then served an apprenticeship as a machinist with Ball & Jewell, Brooklyn, shortly afterward (in 1898) becoming a naturalized citizen of the United States.

After working as a machinist for R. Hoe & Co., the Progressive Iron Works, Chelsea Fibre Mills, and a few small shops, he entered the employ of the Eberhard Faber Pencil Company in 1907. He was a machinist for the company for about three years and foreman for two and a half years prior to becoming master mechanic. He did much to develop the products of the company, and held patents on several inventions of his own, including a mechanical pencil and a pencil lengthener.

Mr. Robinson became an associate-member of the A.S.M.E. in 1915, was president of the Bayview Park Association, Port Washington, L. I., at the time of his death, and belonged to the masonic fraternity.

VERNON ROYLE (1846–1934)

Vernon Royle, president and treasurer since 1885 of John Royle & Sons, machinery manufacturers of Paterson, N. J., died of pneumonia at his home in that city on December 17, 1934. Four distinct lines of industry bear the impression of his inventive and manufacturing ability—photo-engraving, textile and fire-hose weaving, rubber goods, and electrical insulation.

Mr. Royle was born in Paterson on June 9, 1846, the eldest son of John Royle, Sr., of English ancestry, who founded the Royle machinery business in Paterson in 1855. His mother was Agnes Houston Royle, a native of Scotland. His early education was acquired in the "pay schools" of Paterson, public schools not having been established there until about 1858.

He served an apprenticeship in patternmaking with Wm. G. & J. Watson, machinists of Paterson, from 1864 to 1868. He then secured a position in the engravers' joining business of Heber Wells, later Vanderburg & Wells, in New York, N. Y. This contact with the boxwood engraving trade and with the early experiments in photo-engraving led to his later improvements in machinery for that industry.

In 1877 Mr. Royle decided to join the Paterson machine business. He became president when his father retired eight years later and under his guidance and subsisting almost wholly upon his own inventions, the Royle machine shop of the seventies grew to a modern manufacturing plant doing a business that was international in its scope, and the leader in each of its numerous lines. It remained a partnership until 1898, when it was incorporated under the same name of John Royle & Sons. Mr. Royle continued active in the management of the business until his death.

In photo-engraving and related processes, Vernon Royle's machinery for routing, lining, beveling, trimming, and blocking plates for the printing press has materially furthered the development of the world's illustrative arts during the past fifty years. Substantially all commercial weaving of figured textile goods is done today through the medium of jacquard pattern cards made on Royle card-cutting machines. Woven fire hose and gasoline service station hose are productions of Royle circular looms.

In the rubber industry, automobile tire treads and inner tubes are made in vast quantities on Royle tubing machines. The same is true of miscellaneous tubing, garden hose, pencil tips, rubber heels, and hundreds of other articles in soft and hard rubber and celluloid. Royle machinery for applying rubber insulation upon electric wire and cable is likewise the standard process of manufacture.

Mr. Royle was keenly interested in his community. He was secretary of the Paterson Board of Education from 1872 to 1889 and for several terms was a member of the board. He had been a director of the Hamilton Trust Company of Paterson for 25 years, was a director of the Cedar Lawn Cemetery Association, and a member of the local Taxpayers Association. He was a former member of the Passaic Valley Sewage Commission.

Mr. Royle became a member of the A.S.M.E. in 1907 and also belonged to the New Jersey Historical Society, the masonic fraternity, and the Hamilton Club, of Paterson.

He was married in 1872 to Miss Jeannie Malcolm, who died in 1908. He is survived by a son, Vernon E. Royle, now president of the company, and by three grandchildren and two great grandchildren.

GEORGE NICHOLAS SAEGMULLER (1847–1934)

George Nicholas Saegmuller, formerly a member of the firm of Bausch & Lomb Optical Co., Rochester, N. Y., died at his home in Clarendon, Va., on February 12, 1934. He was born just 87 years before, on February 12, 1847, at Neustadt-an-der-Aisch, Bavaria, the son of J. L. and Babette (Bertholdt) Saegmuller.

After four years' training at the polytechnic school at Nuremberg, Mr. Saegmuller went to England, where he was employed as draftsman for four years, 1865–1869, by Cooke & Sons, York, manufacturers of astronomical instruments. The following year he served in the German Army as a one-year volunteer and from 1870 until 1905 was a partner in the firm of Fauth & Co., Washington, D. C., manufacturers of astronomical and engineering instruments. This firm combined with the Bausch & Lomb Optical Co. in 1905 and Mr. Saegmuller held the office of vice-president of the latter until he retired from business in 1925.

In Who's Who in Engineering for 1931 will be found the following statement regarding Mr. Saegmuller's achievements: "Among engineers (he was) best known as the inventor of the solar attachment by means of which the true meridian is found by a single observation of the sun and which is largely used in the surveys of public lands. In astronomical circles he is known as the inventor of the Star Dials which make circles on equatorially mounted telescopes unnecessary and by means of which a star may be found without computing the hour angle; also for accurate graduations produced on automatic dividing engine. For the last thirty years his most important work was in connection with the developments of fire control instruments for the American navy. He, with the then Captain Samson, originated for our navy the first telescopic gunsights, boresights, and in fact all fire control instruments used in our navy and most important of all the development of our large rangefinders which are found in every turret of our battleships."

Mr. Saegmuller became a member of the A.S.M.E. in 1911. He also belonged to The Franklin Institute, the American Association for the Advancement of Science, and several clubs.

Mr. Saegmuller married Maria J. Vandenbergh, in Washington, D. C., in 1874, and is survived by her and by their three sons, J. L., Fred B., and George M. Saegmuller.

EDWIN JOEL SANDERSON (1877–1934)

Edwin Joel Sanderson, manager of the A. C. Ramsey Co., an insurance firm of Terrell, Texas, since 1934, die in Dallas, Texas, on December 27, 1934. He was born at New Vienna, Ohio, on February 18, 1877, the son of Francis Wyatt and Frances B. (West) Sanderson. After a year at the Ohio State University he entered the employ of James Leffel and Company, Springfield, Ohio, manufacturers of water wheel turbines. He remained with this company until 1931, working as draftsman for five years, the latter part of the time in charge of the department, then eight years in the engineering department, four years as sales engineer, twelve as sales manager, and the remainder of the time as general manager of the turbine department. He was manager of the Clark Screw Machine Company, Springfield, for a short time before going to Texas.

Mr. Sanderson became a member of the A.S.M.E. in 1930. He had been secretary-treasurer of the Terrell Building and Loan Association since 1932, had served as a director of the Rotary Club in both Springfield and Terrell and was chairman of the club's Crippled Children's Committee. He was a 33d degree Mason and had been a director of the Chamber of Commerce in Springfield and Terrell. He is survived by his widow, Carrie D. (Van Loan) Sanderson, whom he married in 1932.

NORMAN ROBERT SEIDLE (1877–1934)

Norman Robert Seidle, whose death occurred in Chicago, Ill., on May 14, 1934, was born in Lebanon, Pa., on January 25, 1877, the son of Christian and Jane (Mull) Seidle. He studied at Franklin and Marshall College for two years and at Swarthmore College for one year, then entered the employ of the Standard Boiler Works at Lebanon. With the exception of about a year (1900–1901) spent with the North Penn Iron Company, Philadelphia, he was connected with the Standard Boiler Works until 1904, rising to the position of superintendent and assistant manager, in charge of steel-plate construction.

From 1904 to 1919 Mr. Seidle was manager of the Plate Construction Department of Joseph T. Ryerson & Son, Chicago. He had charge of the design and manufacture of boiler specialties, and of steel pipe lines for hydroelectric developments and water supply, as well as of general plate construction and erection and pressed steel work.

After leaving the Ryerson company Mr. Seidle spent three years as general manager of James G. Heggie & Sons Co., Joliet, Ill.; about a year in the same capacity with the General Boiler Company, Waukegan, Ill.; a similar period as manager of the boiler department of the Harrisburg (Pa.) Manufacturing & Boiler Co.; seven years as treasurer and general manager of the McAleenan Bros. Co., and vice-president of its successor, the McAleenan Corporation, Pittsburgh, Pa.; and two years as vice-president and general manager of the Youngstown Boiler & Tank Co. In December, 1933, he became executive director of the National Steel Tank Association and secretary-treasurer of the Code Authority of the metal tank industry, the position he held at the time of his death.

During the World War Mr. Seidle was a civilian member of the Procurement Division of the United States Army, and a "Minute Man." He became an associate of the A.S.M.E. in 1917, was a past-secretary of the Joliet Rotary Club, and belonged to the Delta Tau Delta fraternity and the Odd Fellows. He was deeply interested in religion and for many years had served as a vestryman in the Episcopal Church.

Mr. Seidle is survived by two sons, N. Robert and Charles A. Seidle. His wife, Florence (Wedekind) Seidle, whom he married in 1905, died in 1926.

GEORGE HOTCHKISS SMITH (1851–1934)

George Hotchkiss Smith, whose death occurred in Providence, R. I., on June 7, 1934, had been a member of the A.S.M.E. since 1881. The fifty-year medal of the Society was presented to him in 1931.

Mr. Smith was born in New Britain, Conn., on December 22, 1851. He supplemented his public schooling with study at Cooper Union, New York, and served an apprenticeship in the machine shop of Smith & Garvin, at Bricksburg (now Lakewood), N. Y. In 1878 he went to Providence, to work as a draftsman for the Brown & Sharpe Manufacturing Co. He was made chief draftsman the following year, and remained in that position until 1885, when he entered business with Elmer A. Beaman under the firm name of the Beaman & Smith Co. He retired from business about the year 1915 because of ill health.

Mr. Smith was a past-president of the Providence Engineering Society and a member of the masonic fraternity. He is survived by a son, Walter J. B. Smith, of Providence. His wife, Louise J. (Luther) Smith, whom he married in 1882, died in 1916.

NATHANIEL AUSTIN SMITHWICK (1864–1934)

Nathaniel Austin Smithwick died on July 6, 1934, at his summer home, Waites Landing, Falmouth Foreside, near Portland, Maine. He had made his home in Maine since his retirement in 1912.

Mr. Smithwick was born at Damariscotta, Me., on March 25, 1864, the son of Captain Frank and Caroline (Austin) Smithwick, and as a boy he sailed with his father to many ports. He attended Lincoln Academy, Newcastle, Me., and Phillips Exeter Academy, Exeter, N. H., spent six months as an apprentice machinist with Barrett Bros., Boston, and was graduated from Worcester Polytechnic Institute, with a B.S. degree, in 1886.

After his graduation Mr. Smithwick spent a year as machinist at the Thomson-Houston Electric Company, at Lynn, Mass., and two years as draftsman for Erastus Woodward, mechanical engineer of Boston. The remainder of his engineering work was done for the Goulds Manufacturing Company, of Seneca Falls, N. Y. He was draftsman for five years, manager of the designing and experimental department for eleven years, and after that chief engineer of the company. He was largely instrumental in introducing triplex power pumps into mining, oil refining, water works, and other fields. He took part in a series of efficiency tests of motor-driven triplex pumps during his early days with the company, and later designed and supervised the installation, at Austin, Texas, of the first electrically operated water works pumping plant in the United States. In 1910 he went to South Africa to further the use of electric power pumps in mining operations. He held patents for governor valves for automatic pumps and for an improvement in vertical suction pumps.

Mr. Smithwick became a member of the A.S.M.E. in 1916 and was always keenly interested in scientific and engineering developments. Yachting and yacht designing were naturally of special interest to him, in view of his early trips with his father.

Surviving Mr. Smithwick are his widow, Mabel (Kenney) Smithwick, whom he married in 1901, and their only son, Austin K. Smithwick.

THOMAS WADE STONE (1877–1933)

Thomas Wade Stone, vice-president and general manager of The Western Gas Construction Company, Ft. Wayne, Ind., died on September 28, 1933, at the Methodist Hospital in that city, a few hours after suffering a stroke of apoplexy.

Mr. Stone was born on October 4, 1877, at New Corydon, Ind., a son of Dr. Michael and Mary (Elzey) Stone. The family moved to New Bremen, Ohio, when he was a small child and he attended public schools there and was graduated from the New Bremen High School in 1894. He then taught school for two years at Kettlersville, Ohio, a small town near New Bremen, and for one year at New Bremen. He attended Armour Institute at Chicago for a year and then entered Ohio State University at Columbus, from which he was graduated in 1902 with a degree in mechanical engineering. He won honors in mathematics and was elected to membership in Sigma Xi, honorary scientific fraternity.

Following his graduation from the university Mr. Stone went to Fort Wayne to become draftsman for The Western Gas Construction Company. He was advanced to the position of chief draftsman the following year, became chief engineer in 1916, and was made vice-president of the firm in 1928. In 1931 he also became general manager of the plant. Patent office records dating back to 1914 show the result of Mr. Stone's inventive ability in the gas manufacturing industry. The inventions credited to him include various mechanisms for use in the manufacture of gas, purifying devices, and other apparatus for the industry.

Mr. Stone became a junior member of the A.S.M.E. in 1903 and a member in 1916. He also belonged to the American, Canadian, and Indiana Gas Associations and to a number of clubs in Ft. Wayne, in many of whose civic activities he was actively interested. He belonged to several of the higher orders in the masonic fraternity. During the World War he aided in the manufacture of phenol at the plant of The Western Gas Construction Company for use in making explosives. He participated in numerous drives for the sale of Liberty Loan bonds, raising money for the Red Cross, and other war activities.

Mr. Stone was married to Miss Dorothy M. Greiwe, of New Bremen, in 1905, and is survived by her and by two sisters.

EDWIN STRUCKMANN (1872–1934)

Edwin Struckmann, since 1928 connected with the Lone Star Portland Cement Company of Kansas, died in Kansas City, Mo., on March 15, 1934. He was born at Aalborg, Denmark, on April 24, 1872, the son of a Danish sea captain. He served a four-year apprenticeship at the Nielsen & Winther Tool & Machinery Co., Copenhagen, and worked for some months as a mechanic at Kofoed & Hauberg's Machine Co. in that city. He also passed examinations for engineering service in the Danish Mercantile Marine and was a reserve assistant engineer in the Danish Royal Navy for about seven months.

His desire to become a navigating officer, however, was not to be fulfilled. On the return from a trip before the mast to South America his eyesight proved defective. He turned to engineering with the hope of becoming a ship's engineer, but after his graduation in 1896 from the Technical Institute in Copenhagen, he became interested in brick plants, and entered the employ of F. L. Smidth & Co., of Copenhagen. He was connected with this company for ten years, working in various capacities from draftsman to engineer in charge of construction and operation of brick and cement plants. One of his assignments was the construction of a brick plant in the northern part of Sweden, where the region was so desolate that a railroad and town had to be built before work on the plant itself could be started. Later he was sent to Russia to reconstruct and operate a brick plant and was there during the Russo-Japanese War, suffering many of the hardships of the times. A daughter, born to Mr. and Mrs. Struckmann in Sweden, died while they were in Russia. His final assignment with the company was the construction of a plant for the Canadian Portland Cement Company at Port Colborne, Ontario, Canada.

Since 1907 Mr. Struckmann had been connected with a number of portland cement companies in the United States. For something over three years he was superintendent of the Kansas City Portland Cement Company's plant and from then until the fall of 1911 held a similar position with the Continental Portland Cement Company of St. Louis, Mo. He then went to Des Moines, Iowa, where he was superintendent of the Iowa Portland Cement Company's plant until 1920 and then was connected with the Pyramid Portland Cement Company, first as vice-president and general manager, and from 1922 to 1925 as president and general manager.

From 1925 to 1929 he was employed by the Knickerbocker Portland Cement Company, Hudson, N. Y., first as superintendent, then as general superintendent. In 1929 he became general superintendent of the Western Division of the Lone Star Cement Company of Kansas, at Kansas City, Mo. At the time of his death he was stationed at Bonner Springs, Kan.

Mr. Struckmann became a member of the A.S.M.E. in 1913, and belonged also to the American Society of Danish Engineers. He is survived by his widow, Marie V. (Olsen) Struckmann, whom he married in 1900, and by a son, Holger Struckmann, as well as by a sister, Mrs. Holden Knudsen, of Copenhagen, Denmark.

CHARLES ANTHONY SUPIOT (1906–1933)

Charles Anthony Supiot, mechanical engineer for Hires, Castner & Harris, Inc., Philadelphia, Pa., died of pneumonia on May 21, 1933.

Mt. Supiot was born at Westtown, Pa., on October 6, 1906, the son of Charles Aime and Cecilia Supiot. He was graduated from the University of Pennsylvania in 1929 with a B.S. degree in mechanical engineering. He had been connected with Hires, Castner & Harris, Inc., since his graduation, beginning as a draftsman and subsequently becoming mechanical engineer for the company.

Mr. Supiot was a member of the A.S.M.E. Student Branch at the University of Pennsylvania and became a junior member of the Society upon graduation. He is survived by his mother; his father died in November, 1933.

CARROLL D. THOMAS (1863–1934)

Carroll D. Thomas, for nearly fifteen years chief boiler inspector for the State of Oregon, died at St. Vincent's Hospital in Portland, Ore., on December 24, 1934. He was born on November 13, 1863, at Hollansburg, Ohio, where he spent his early life. He attended the Central Normal College at Danville, Ind., for two years, and taught school and was principal of public schools in Ohio and Indiana for a number of years. He went to Oregon in 1884 and taught school in that state the following two years. He joined the gold rush to Alaska during 1887 and became operating engineer for the Treadwell Gold Mining Company, of Juneau. He returned to the States in 1888, engaging in mining operations in Colorado. In 1890 he went to Unionville, Mont., where he worked for a year on erection for the McCann Gold Mining Company, and from then through 1893 as designing, erecting, and operating engineer for the Whitlatch Union Gold Mining Company. For ten years beginning in 1894 he was engaged in erection work in Colorado mines, part of the time in the Cripple Creek district and from 1898 on as chief operating engineer of the Portland Gold Mining Company at Victor, Colo.

Mr. Thomas entered the power field in 1905 as operating engineer for the Pacific Power & Light Co. at Astoria, Ore. Two years later he became boiler inspector for the Hartford (Conn.) Steam Boiler Inspection and Insurance Company, in the Portland, Oregon, district. He continued in this position until he became state boiler inspector in 1921.

Mr. Thomas became an associate-member of the A.S.M.E. in 1925, and was a member of the Conference Committee to the Boiler Code Committee of the Society. He had served as chairman of the National Board of Boiler and Pressure Vessel Inspectors and as technical adviser to the Industrial Accident Commission of the State of Oregon.

Surviving Mr. Thomas are his widow, Mrs. Beatrice Thomas, of Salem, and a son, Carl F. Thomas, of Portland, as well as three brothers and two grandchildren.

OSKAR VON MILLER (1855–1934)

Oskar von Miller, pioneer in electrical generation and distribution in Germany, founder of the world-famous Deutsches Museum, Munich, and honorary member of the A.S.M.E., died in Munich on April 9, 1934. His long and honorable career was closely identified in Germany with the growth of the electrical industry which has wrought far-reaching changes in industrial and domestic life and in western culture. He labored with vision, intelligence, and energy, keenly conscious of these changes and astute enough to realize the necessity of impressing upon the youth of Germany the significance of the powerful technological influences that determine the environment in which their lives are lived.

Oskar von Miller was born at Munich, May 7, 1855, the tenth son in a family of fourteen children. His grandfather had been director in the Academy of Art in Munich, and his father, Ferdinand von Miller, the head of a bronze foundry, was an artizan of considerable ability and won government awards for his statues. His mother, Anna von Pösl, the daughter of a family of high social and governmental standing, whose marriage with Ferdinand von Miller was

contrary to the wishes of her parents, found time, in spite of the large family, to keep the books for the foundry.

The children were strictly brought up and encouraged in self-support at an early age. Oskar von Miller attended the Technische Hochschule at Munich, and, after having been graduated from it at the head of his class, entered the civil service, holding several transportation posts during the next few years. Among his early engineering tasks was the building of a bridge over the River Main in the winter of 1879 when the weather was so cold that no one else could be found to do the work.

But von Miller's major engineering interest, that of the electrical industry, began in 1881 when he asked for and received a furlough and journeyed to Paris to see the First Electrical Exposition. Here he became intensely interested in electricity and its applications with the result that he organized an electrical exhibit in Munich in 1882, at which the French engineer, Deprez, successfully transmitted direct current from Miesbach to the Glaspalast in Munich, a distance of 57 km.

Following the successful Munich exhibit, von Miller was sent abroad to France, England, and America to study the technology of the infant electrical industries. It was on this journey that he met Edison for the first time, and while in Vienna, after his return home, he was invited by Emil Rathenau to join in the introduction of Mr. Edison's electric lighting plants in Germany. He refused this unexpected and flattering offer because of his desire to spend his talents and energy in the service of the Bavarian state. He envisioned the development of the water-power resources of Bavaria, but his superior considered the plans premature. Perhaps it was the spirit of the times, combined with his own enthusiasm and the experiences of his travels that inspired young von Miller in the vision he was to live to see a reality many years later. The young German empire, politically powerful after the crushing defeats of Austria and France, and guided by the forceful Bismark, was emerging from an agricultural into an industrial nation of first rank, rich in resources of materials and human energy, and possessed of the scientific spirit which provides the basis for technological growth. Progress and technological development were in the air.

In Berlin, Emil Rathenau had formed the German Edison Company, now known as the A.E.G., and in 1883 von Miller joined him as a director of the company. By 1884 they had opened the first central station in Berlin. Here the boilers were set above the engine room where Edison bipolar dynamos were driven by belt, three to every engine. For the next five years they were busy developing electrical power plants throughout Germany.

Yielding to his desire to return to his native Munich, von Miller left the A.E.G. in 1889. Life was not easy, but the director of the Portland Cement Works at Lauffen sought him out with a proposition to supply the city of Heilbronn with power from Lauffen. This he accomplished, completing the work just prior to the electrical exposition at Frankfurt, in 1891, of which he became the president.

While power had been transmitted at the Munich exposition, electrical generation was there the major interest. Electrical transmission over considerable distances appealed to von Miller and the idea of transmitting electricity from his Lauffen plant to Frankfurt seemed worth trying. Consultation with Charles E. L. Brown convinced him that an alternating current could be so transmitted, as indeed it was, at 25,000 volts over a three-phase system 178 km in length. Of the 234 horsepower at Lauffen, 181 reached Frankfurt, a loss of only 22.6 per cent. In the same year, at Kassel, von Miller installed the first alternating-direct-current system, in which electricity was transmitted as alternating current and converted to direct current for use.

Following the Frankfurt exhibition, von Miller was firmly established as an electrical engineer and his genius was providing electric lighting and power plants throughout Germany. Space will not permit even a list of these projects. His interests spread to railway electrification and to the use of electricity in the chemical industry and for cooking and domestic appliances. In 1911 there were initiated the plans for the comprehensive hydroelectric power development of Bavaria, totaling 600,000 kva and comprising the famous 168,000-hp project at Walchensee, opened in 1924, with its 1000 km, 100,000-volt transmission system, and a yearly output of 180 million kilowatthours. Thus came to fruition the dream of his young manhood. In 1930, an extensive plan for the electrification of all of Germany engaged his attention.

But important as these engineering achievements of von Miller proved to be, he is best known throughout Germany and the world at large for the Deutsches Museum in Munich. His interest in the museum idea is said to have originated in a visit to Kensington Museum in 1878 when he saw the famous "Puffing Billy," built by William Hedley in 1813, and Stephenson's "Rocket." Lying fallow in his mind for a quarter of a century, the idea of a museum for his native city in which laymen and young people could see the industrial foundations on which our present civilization is based was presented by him to a group of government representatives, fellow citizens, scholars, and technologists in 1903, and he became the founder, president, and director of the institution. To this project he gave the best of his energy and enthusiasm and drove it through to an amazingly popular success in spite of tremendous difficulties.

When it outgrew its original quarters a few years after its opening, the city of Munich provided a commanding site for new buildings on a beautifully located island in the River Isar. Here, in 1925, after the long delay of the War, the present museum was opened to the public. Thousands of visitors to Munich pass through its 400 separate rooms and halls and from a city of 700,000, more than a million pass every year along the nine miles of its exhibits. It has provided inspiration for the founding of similar museums in this country. Space will not permit an estimate of what this great contribution means to the world's understanding of the conditions under which men live and how technology is constantly undergoing development.

Mrs. von Miller, whose death in the fall of 1933 broke a family group unusual in its solidarity, was Marie Seitz, the daughter of Franz Seitz, a professor at the University of Munich. Of her marriage to Dr. von Miller in 1884 seven children were born, three sons and four daughters, two of whom died at an early age. The family was accustomed to meet for a day or an evening together every two weeks, and each year gathered for a religious service at the little church in Neuhausen where Dr. and Mrs. von Miller were married.

Von Miller possessed to a rare degree human qualities that endeared him to every one. His tireless energy permitted him to accomplish great deeds and his infectious enthusiasm carried others along with him in the pursuit of his objectives. Breadth of vision provided these objectives in heroic measure, and a keen sense of his obligation to his country and its culture led him to give his services to such projects of social significance as the electrical exhibitions of his early manhood and the museum and the electrical development of Bavaria which occupied his later years. When the great von Miller saw a need, he did not stop until he had satisfied it, true to his family motto, "If I rest, I rust."

JOSEPH ADDISON WADDELL, JR. (1877–1934)

Joseph Addison Waddell, Jr., western district manager of the Spencer Heater Company, a subsidiary of the Cord Corporation, with offices in Chicago, Ill., died suddenly of coronary thrombosis on April 28, 1934, while in St. Louis, Mo.

The son of Legh Richmond Waddell and nephew of Senator Joseph Addison Waddell, of Virginia, he was born at Staunton, Va., on November 5, 1877. He received a B.S. degree in mechanical engineering in 1900 and M.E. degree two years later from Virginia Polytechnic Institute. While working for his Master's degree he taught mathematics and experimental engineering at the Institute, and during the next year was employed in the erecting and layout department of the Deane Steam Pump Company, Holyoke, Mass.

Mr. Waddell first became associated with the Spencer Heater Company, then located in Scranton, Pa., in 1903 as engineer in charge of sales and mechanical engineering. Two years later he was made general superintendent and in 1913 works manager, with entire charge of production. He held this post until 1917, when he was released to serve in the Production Division of the Emergency Fleet Corporation, United States Shipping Board. He had charge of the Boiler Section and was responsible for the allocation of material to the various manufacturing companies throughout the United States which were building boilers for the Corporation. After the armistice he adjusted contracts and claims arising from cancelations of orders, remaining with the Corporation until the end of 1919.

During the next four years Mr. Waddell was connected with Morris and Company, of Chicago, as manager of its Phosphate Mining Department at Bartow, Fla. He directed the construction of a hydraulic plant, including the power plant, washing and separating plants, and drying and storage plants, repair shop, and other buildings, and installed the machinery and other equipment. After the completion of the plant he directed the operation of the mine and the production of pebble phosphate rock until the company was purchased by Armour and Company, Chicago. A model of the Bartow phosphate mine has been placed in the Smithsonian Institution in Washington, D. C.

From then until 1927 he was not engaged in engineering work, and in 1924 he resigned his membership in the A.S.M.E., which he had held since 1913. He applied for reinstatement in 1930.

When Mr. Waddell again took up engineering duties in 1927 it was with the Spencer Heater Company, located at Williamsport, Pa., in the capacity of chief engineer. He held patents on Spencer heaters dating back to as early as 1910 and after his return to the company

developed the present line of products and was granted patents in both the United States and Canada on magazine stoking boilers for heating purposes. Since 1933 he had been stationed at Chicago as western district manager, but had also continued as chief engineer of the company.

Surviving Mr. Waddell are his widow; a daughter, Martha; and a son, Joseph Addison Waddell, 3rd, also with the Spencer Heater Company.

VINCENT L. WALTERS (1868–1934)

Vincent L. Walters, who became an associate-member of the A.S.M.E. in 1920, died on April 30, 1934. He was born at Bilston, England, on February 6, 1868, and was educated in that country. His father was master mechanic in a large steel mill and the son trained for erecting and mill work. He had not worked long in England, however, when his brother, in the United States, advised him to come to this country. He wished for a position as draftsman, but there were so many applicants for such work in New York that for a number of years he found erecting and mill work offered better pay.

Later, however, he obtained work in which he was more interested as designer and experimentalist for the Ideal Cash Register Company, Bound Brook, N. J. He was connected with this company until 1901, designing, among other things, a spring winding machine and special dies, and having charge of the tool and gage room.

From 1901 to 1912 Mr. Walters was employed by the Weston Electrical Instrument Company, Newark, N. J. He designed intricate mechanisms for a variety of instruments and machines and for three years was in charge of the tool department of the company.

After a few months in 1912 and 1913 as experimentalist and model maker for Sloan and Chase, Newark, Mr. Walters went to Orange, N. J., for the manufacture of the Monroe calculating machine, on the experimental stages of which he had been working for some time. He designed all the tools necessary for its manufacture and was assistant superintendent of the Monroe Calculating Machine Company for four years.

Mr. Walters next spent several months with the Mehl Machine Tool & Die Co., Roselle, N. J., then became superintendent of A. Rimelspacher & Co., designing a special thread rolling machine and directing the manufacture of machinery and tools. In the summer of 1917 his aid was solicited by Joseph Gough, of Newark, and during the following winter he worked on the perfection of a universal joint gage for a Government assignment. Subsequently he became foreman for Jackson Waddell, Newark, where he remained until the latter part of 1919. He then became general foreman for the Pierce Accounting Machine Company, Inc., New York, for whom he carried on experimental as well as manufacturing work until 1924.

FRANK ROSE WHEELER (1878–1934)

Frank Rose Wheeler, a specialist in condensing equipment, died of a cerebral hemorrhage at his home in Chicago, Ill., on March 1, 1934.

Mr. Wheeler was born at Wheeler, N. Y., on April 1, 1878, a son of Don D. and Mary (Rose) Wheeler. He was graduated from high school in Tacoma, Wash., and subsequently was a special student at Leland Stanford Jr. University in 1898 and 1899 and from 1901 to 1903. During these years he also worked as a draftsman for the City of Tacoma and the Northern Pacific Railway, and following his graduation, for C. C. Moore & Co., of Los Angeles and San Francisco, and as engineer and vice-president of the Tracy Engineering Company, Los Angeles, working upon the design of power plants.

In 1904 Mr. Wheeler became Pacific Coast manager for the C. H. Wheeler Mfg. Co., of Philadelphia, in charge of the sale of condensing apparatus. He took over the Chicago district in 1916 and remained with the company until 1929, when he became manager of the Condenser Heater Department of the Elliott Company, Jeannette, Pa. He spent three years in this position, and during the remainder of his life was connected with the Utilities Power & Light Co. in Chicago.

Mr. Wheeler became a member of the A.S.M.E. in 1909 and also belonged to the Western Society of Engineers, the Beta Theta Pi fraternity, and to several clubs in Chicago and San Francisco. He was a contributor to the technical press.

Mr. Wheeler is survived by his widow, Blanche (Harper) Wheeler, whom he married in Pittsburgh in 1912.

GEORGE E. WHIPPLE (1890–1934)

George E. Whipple, solicitor for United Drydocks, Inc., New York, N. Y., died of a heart attack at his home in Teaneck, N. J., on May 3, 1934. He was born in Brooklyn, N. Y., on January 3, 1890, son of George E. and Nellie T. (Doody) Whipple. He attended public schools in Brooklyn and studied art in New York. The most of his engineering experience was with Theo. A. Crane's Sons Co., of Brooklyn, with which he became associated in 1917. From then until 1928 he was located in the New York office of the company, as draftsman and designer, and assistant executive. He was also assistant superintendent of the shipyard, supervising all repairs on boilers, engines, and hulls. He became solicitor for ship repairs for the company in 1929 and the following year took the position of solicitor with United Drydocks, Inc. He was a licensed ship engineer, first class.

Mr. Whipple became an associate-member of the A.S.M.E. in 1927. He was a 32d degree Mason and belonged to the Sons of the American Revolution, Odd Fellows, and Rosicrucians. He was a deputy warden of the New Jersey Fish and Game Commission. In addition painting in oils and sketching in pen and ink, he was interested in photography and in floral horticulture.

Mr. Whipple is survived by his widow, Marie B. (Arden) Whipple, whom he married in 1920.

WILLIAM HIGH WILLISTON (1879–1934)

William High Williston was born on October 10, 1879, at Somerville, Mass., where he secured his grammar and high-school education. He was graduated from the Massachusetts Institute of Technology in 1902 with a B.S. degree, and was employed continuously from December of that year until his death on June 7, 1934, by Manning, Maxwell & Moore, Inc. He spent his early years with the company in drafting and designing at the Hancock Inspirator Company and Hayden & Derby Manufacturing Co., rising to the position of chief designer. He became mechanical assistant to the vice-president of the parent company in 1913 and was made assistant works manager of the Boston plant three years later. Subsequently he was promoted to the position of works manager of that plant, then railroad sales manager. He was appointed vice-president in charge of railroad sales in 1927, and at the time of his death was director and vice-president in charge of railroad sales of the subsidiary company, Consolidated Ashcroft Hancock Co., Inc., as well as works manager of the Boston plant.

Mr. Williston became a member of the A.S.M.E. in 1925. He was an enthusiastic philatelist. He is survived by his widow, a son and a daughter, and his father, Belvin T. Williston.

WALTER WOOD (1849–1934)

Walter Wood, of R. D. Wood & Co., Philadelphia, Pa., manufacturers of cast-iron water and gas pipe, died on April 20, 1934, in Washington, D. C., while there on a business trip. He was born in Philadelphia on December 6, 1849, a son of Julianna (Randolph) Wood and of Richard Davis Wood, who was a partner in the firm of R. D. Wood & Co., and one of the organizers of the Pennsylvania Railroad.

Mr. Wood was graduated from Haverford College in 1867 with the degree of Bachelor of Arts and took post-graduate work at Harvard University, securing his A.B. degree there in 1868. He then began work in the office of his father, becoming managing partner when his father died in 1869. He was also president of the Florence (N. J.) Pipe Foundry & Machine Co., owner of The Millville (N. J.) Daily Republican, and treasurer of the Cumberland County Gas Company, the Millville Electric Light Company, the Millville Water Company, and the High Pressure Supply Company.

Mr. Wood also interested himself in civic affairs. He became a member of the Committee of One Hundred (Philadelphia) in 1880, and served for four years with this group, which investigated and attempted to improve local governmental conditions. From 1887 to 1895 he served as a Civil Service examiner and was a trustee of Haverford College. In recent years he had been an active proponent of the plan to build a ship canal across New Jersey, linking the Delaware River at Bordentown with Raritan Bay. He advocated the project before the House Rivers and Harbors Committee a few weeks prior to his death.

Mr. Wood became an associate of the A.S.M.E. in 1880 and a member ten years later. He was one of the organizers and charter members of the American Foundrymen's Association and a member of the American Institute of Mining and Metallurgical Engineers, American Society for Testing Materials, American Waterworks Association, and of a number of clubs in Philadelphia, New York, Washington, D. C., and Boston. He was a director of the Philadelphia Bourse and of the Burlington City Loan & Trust Co. He was unmarried and is survived by six nieces, and by two nephews, Richard and Grahame Wood, all of Philadelphia.

EDMUND WILHELM ZEH (1867–1934)

Edmund Wilhelm Zeh, president of the Zeh & Hahnemann Co., Newark, N. J., manufacturers of presses, dies, and automatic machinery for working sheet metals, died on March 17, 1934.

Mr. Zeh was a native of Germany, having been born at Frankfort-on-Main on September 12, 1867, the son of Carl Wilhelm and Catharina (Emmel) Zeh. At the age of 16 he entered the shops of Collet & Engelhard, at Offenbach-on-Main and after two years there spent four more in securing drafting-room practice with the company. He then entered the technical high school at Darmstadt, from which he was graduated in 1891 with an M.E. degree. He returned to the employ of Collet & Engelhard as a constructor and continued in that work until 1893.

Mr. Zeh came to the United States in 1893 to see the World's Fair at Chicago and remained to work as a designer for the Niagara Machine & Tool Works, in Buffalo, N. Y. He was made superintendent of the works the following year, and held that position until 1904, when he helped to found the Zeh & Hahnemann Co. He held patents on percussion power presses, details for double crank presses, double action presses, stacking devices, guards, button machinery, and can body machinery.

Mr. Zeh became a member of the A.S.M.E. in 1915. He was president of the Newark Technical Society and a member of the Verein deutscher Ingenieure. He is survived by his widow, Anna M. (Mantz) Zeh, whom he married in 1889, and by one son, now president of the Zeh & Hahnemann Co.

The Design and Performance of a High-Pressure Axial-Flow Fan

By LIONEL S. MARKS[1] AND THOMAS FLINT,[2] CAMBRIDGE, MASS.

This paper deals with the design and performance of an eight-bladed axial-flow fan to give high pressures and an approximately constant horsepower characteristic. The performance in the laboratory indicates the attainment of a static pressure of 22.2 in. of water at a peak efficiency of 77.5 per cent, with a peripheral speed of 40,000 fpm. The no-flow horsepower is only 16 per cent greater than at peak efficiency. The performance is compared with that predictable from wind-tunnel tests of a cascaded series of air foils.

A PAPER[3] published in the A.S.M.E. Transactions in 1934 discussed the design of a three-bladed axial-flow fan and gave results of tests conducted with it. This fan gave a peak efficiency of 80 per cent and could develop a static pressure (according to tests at 18,000 fpm peripheral velocity) of 18.5 in. of water at the permissible peripheral speed of 4000 fpm. There was one feature about the performance of this fan which is shared with many other fans but which was regarded as unsatisfactory, namely, the horsepower output increased steadily as the air flow was restricted until at the no-flow condition it was approximately 125 per cent greater than at peak efficiency.

The present paper deals with an attempt to design a fan (1) for still higher pressures, approximating the practical limits for the axial-flow type and (2) with a horsepower input which does not rise materially above its value at peak efficiency. It was realized that in order to obtain these objectives it would probably be necessary to accept an efficiency lower than that of the three-bladed fan, but this did not seem important as long as the efficiency was high.

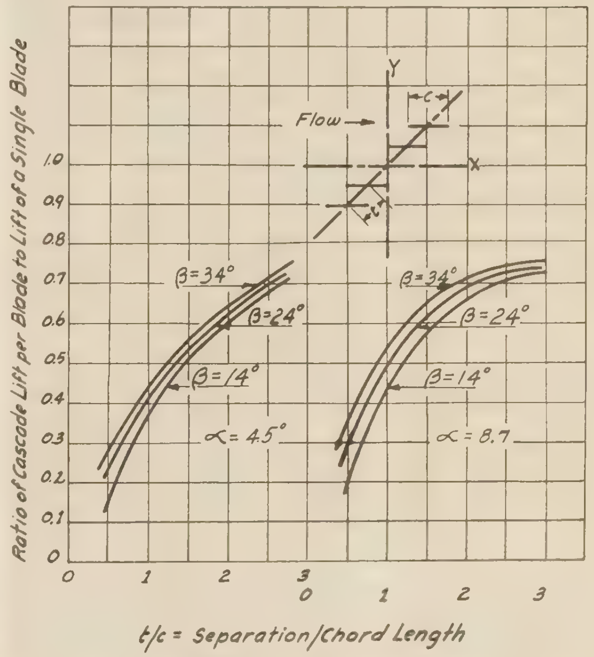

FIG. 1 LIFT COEFFICIENTS FOR A CASCADE SERIES OF AIRFOIL SECTIONS

Various considerations indicate that a larger hub would be necessary and a larger number of blades desirable. It was decided arbitrarily to use eight blades.

Cascade Tests

It was proposed to base the design on airfoil theory and wind-tunnel constants. No wind-tunnel data were found in the literature on the effect of mutual blade interference in a cascaded series of airfoils. To supply this information, S. Ober, Associate Professor of Aeronautical Engineering, Massachusetts Institute of Technology, conducted a series of wind-tunnel tests on a cascade of five identical airfoils with boundary walls at both ends (representing the hub and casing of a fan) and determined the lift and drag of the central airfoil which was free from the boundary walls. The Gottingen No. 429 profile with a straight chord was investigated, and tests were made for various stagger angles β, for two angles of attack α, and for a series of values of the ratio of separation along the line of stagger to chord length t/c. Ratio of the lift observed to the lift on a single blade is shown in Fig. 1.

According to Professor Ober, these tests are neither extensive nor accurate enough to warrant complete confidence. The fact that they are strongly at variance with the theoretical values of Numachi[4] would seem to invalidate Numachi's theory rather than cast doubt on the test results.

[1] Professor of Mechanical Engineering, Harvard University. Mem. A.S.M.E. Professor Marks was born in Birmingham, England. He received the degree of B.Sc. from the University of London in 1892 and M.M.E. from Cornell University in 1894. He was with the Ames Iron Works, Oswego, N. Y., in 1894 and then went to Harvard University as instructor in mechanical engineering. In 1900 he was made assistant professor and in 1909 was advanced to his present position. Professor Marks is the author of "Steam Tables and Diagrams," "Gas and Oil Engines," "Mechanical Engineers' Handbook," and "The Airplane Engine," and has contributed numerous articles to the technical press.

[2] Designer of conveying equipment, J. W. Greer Company, Cambridge, Mass. Jun. A.S.M.E. Mr. Flint was graduated from Harvard University with a B.S. degree in 1928, and received his M.S. degree the following year from the same University. He has held positions as assistant foreman in the cold-rolling department, American Steel & Wire Company, 1929–1930; engineer, lead extrusion department, Simplex Wire & Cable Company, 1930–1933; and assistant in mechanical engineering, Harvard Graduate School of Engineering, 1933–1935.

[3] "The Design and Performance of an Axial-Flow Fan," by L. S. Marks and J. R. Weske, Trans. A.S.M.E., vol. 56, 1934, paper AER-56-13, pp. 807–813.

Contributed by the Aeronautics Division for presentation at the Annual Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS to be held in New York, N. Y., December 2 to 6, 1935.

Discussion of the paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

[4] "Aerofoil Theory of Propeller Turbines and Propeller Pumps With Special Reference to the Effects of Blade Interference Upon the Lift and the Cavitation," by F. Numachi, Tech. Reports, Tohoku Imperial University, Sendai, Japan, vol. 7–8, no. 3, 1927–1928, pp. 411–469.

In view of these uncertainties the proposed basis of design was abandoned. The analysis of the actual fan performance on the basis of the coefficients given in Fig. 1 shows, however, that the cascade-test results cannot be much in error and indicates the desirability of further work of this kind as the basis of axial-flow-fan design.

Design Method Used

The design method decided upon was based on the equations recently presented by O'Brien and Folsom,[5] although, subsequently,

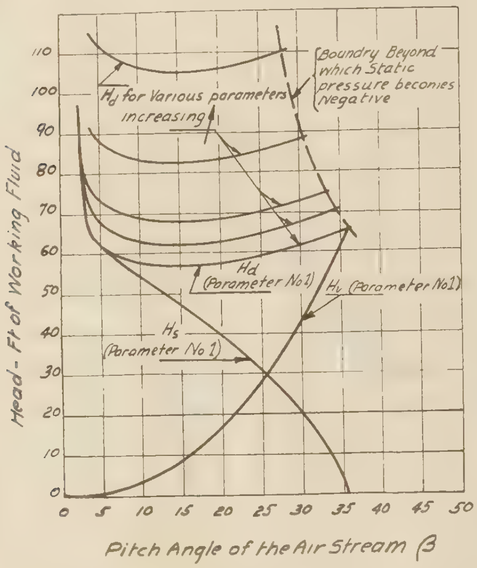

Fig. 2 Variation of Developed Head, Velocity Head, and Static Head With Pitch Angle of the Air Stream for one Particular Value of the Parameters C_L, c/t, and λ as Taken From O'Brien and Folsom's[5] Equations

(U = 100 fps and g = 32.2 ft per sec per sec.)

it was found to be less accurate than the method employing the results of the cascade-tests. In a discussion[6] of the paper[3] on the design and performance of an axial-flow fan, O'Brien and Folsom showed that these equations gave results agreeing satisfactorily with the observed performance. On this somewhat meager basis, it was decided to utilize the O'Brien and Folsom procedure. As stated by them, "The method of computation is essentially that used by Pfleiderer[7] with, however, modifications which result in a better agreement between the computed and the measured head-capacity characteristics."

In these equations, the following symbols are used and apply to any cylindrical section:

H_d = the total or developed head, ft of the working fluid
H_v = velocity head, ft of the working fluid
H_s = static head, ft of the working fluid
U = the tangential velocity of the blade section, fpm
V_{u3} = the tangential velocity of the air at exit, fpm
V_f = the axial velocity of the air at entrance, fpm
C_L = the lift coefficient with infinite aspect ratio
C_D = the drag coefficient with infinite aspect ratio
c = the chord length, ft
t = the circumferential distance between blades, ft
β = the pitch angle of the air stream, or the angle between the geometric mean value of the relative velocities in front of and behind the blade and the direction of blade motion
λ = $\tan^{-1}(C_D/C_L)$.

Combining the equations of O'Brien and Folsom

$$H_d = \frac{2U^2}{g}\,\frac{\left[C_L\,\dfrac{c}{t}\,\dfrac{\sin(\beta+\lambda)}{\cos\lambda}\right]}{\left[4\sin\beta\cos\beta + C_L\,\dfrac{c}{t}\,\dfrac{\sin(\beta+\lambda)}{\cos\lambda}\right]} \quad \text{[1]}$$

$$H_d = H_v + H_s \quad \text{[2]}$$

$$H_v = (V^2_{u3} + V_f^2)/2g \quad \text{[3]}$$

$$V_{u3} = H_d\,g/U \quad \text{[4]}$$

$$V_f = (U - \tfrac{1}{2}V_{u3})\tan\beta \quad \text{[5]}$$

A graphical representation of Equation [1] is shown in Fig. 2, in which H_d is plotted against β. A series of H_d curves is given for constant values of C_L, c/t, and λ. For the lowest H_d curve shown, there are added corresponding velocity-head and static-pressure curves. At values of β greater than that at which the H_d and H_v curves intersect, the static pressure becomes negative. The bounding broken curve on the right gives the values of β at which the static pressure becomes zero.

The velocity of the air stream at entrance V_f is assumed constant at every cylindrical section. As the tangential velocity of the blade section varies with the radius, the pitch angle of the air stream must increase from blade tip to hub as seen from Fig. 3. Fig. 2 shows that the static pressure diminishes continuously to zero as β is increased. The greater the combined value of the parameter determining H_d, or the greater C_L, c/t, and λ, the sooner this effect occurs. In order to keep H_d constant at every cylindrical section and thus avoid losses from re-

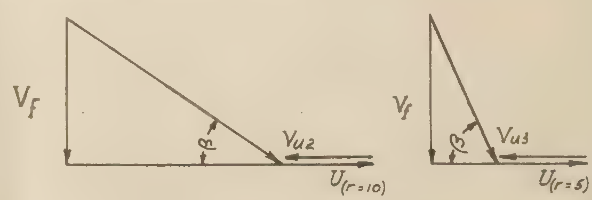

Fig. 3 Variation of Pitch Angle of the Air Stream With Tangential Velocity of Airfoil

circulation through the fan, it is necessary to increase C_L and c/t. It is obvious, therefore, that for efficient high-pressure operation a large hub must be used.

There are three ways by which the head developed H_d may be maintained as the tangential velocity decreases toward the hub:

1 By choosing increasingly thick or increasingly curved profiles. This will increase somewhat the value of C_L with an attendant slight increase in C_D.

2 By increasing the angle of attack α. This angle is the difference between β and the pitch angle of the blade section, at any cylindrical section. This method increases the value of C_L rapidly but soon begins to increase C_D at an even greater rate, as shown in Fig. 4.

3 By increasing the length of the blade-section chord. This method produces a reasonable increase in lift without excessive

[5] "Propeller Pumps," by M. P. O'Brien and R. G. Folsom, Trans. A.S.M.E., vol. 57, July, 1935, paper HYD-57-3, pp. 197–202.

[6] Discussion by M. P. O'Brien and R. G. Folsom of "The Design and Performance of an Axial-Flow Fan," by L. S. Marks and J. R. Weske, Trans. A.S.M.E., vol. 57, August, 1935, p. 344.

[7] "Die Kreiselpumpen," by Carl Pfleiderer, Julius Springer, Berlin, 1932.

increase in drag and was the method selected for the design of this fan. Another important reason for choosing this method in preference to the conventional method (increasing the angle of attack) was that it rendered it possible to design the fan so that no blade section had a pitch angle greater than the angle of attack at which that section would "burble" or break down. It was hoped that the elimination of the break-down point, with its attendant increase in drag, would effectively prevent the rise of horsepower input at low capacity.

After selecting a convenient speed for the fan design, an approach velocity V_f was selected so as to give reasonable capacity. The profile section was chosen from those listed in the N.A.C.A.

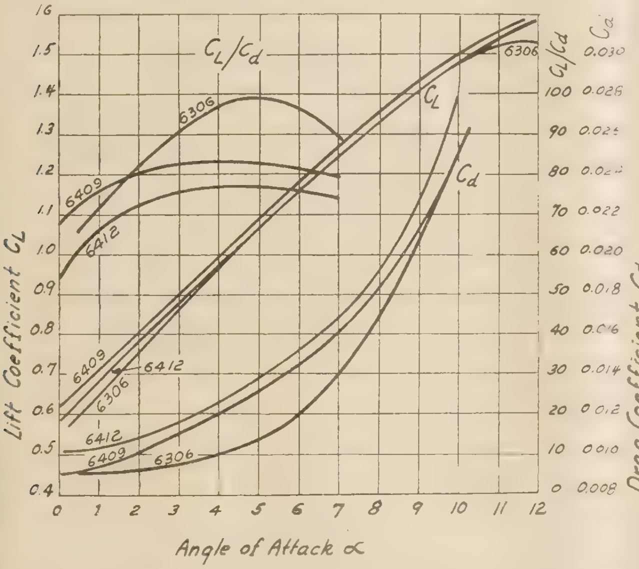

Fig. 4 Properties of the N.A.C.A. Profiles Used for the Axial-Flow Fan

Report No. 460, having in mind high efficiency (high lift-drag ratio), a high value of the angle of attack at the burbling point, and adequate strength for the inside section. The properties of the three sections selected are shown in Fig. 4.

The outside section was designed first, the outside diameter of the fan being determined by existing laboratory equipment. At each section, the angle of attack used was that for maximum lift-drag ratio at the design point of the fan. The value of t for any cylindrical section was determined by the number of blades. From this point on, a cut-and-try procedure was followed. A value of H_d was chosen somewhat higher than the desired static pressure. The tangential velocity of the air at exit V_{u3} was found from Equation [4], β from Equation [5], H_s from Equation [3], and H_s from Equation [2]. A weighted average, based on the square of the radius of each section designed, was taken of the static heads developed at the three sections designed. These were (1) the outside section with a radius of 9.65 in., (2) the section next to the hub with a radius of 6.76 in., and (3) a section midway between these two with a radius of 8.2 in. If this average gave the desired static head, the original H_d was used; if not, a new calculation was made. When the proper value for H_d was found, Equation [1] was solved for c for each section. The hub diameter was taken as that at which the pitch angle of the blade exceeded the angle of attack at which burble occurs.

The section at the hub and the center section were designed in the same way, keeping H_d and V_f constant for all sections.

Each of the static heads resulting was corrected for section-drag loss H_p, which equals $C_L(c/t)(V^2/2g)$ $(\tan\lambda/\sin\beta)$, where $V^2 = V_f^2 + (U - \frac{1}{2}V_{u3})^2$.

No correction was made for losses from friction in the air passages because of the shortness of the passages.

The details of the fan design are given in the pattern drawing, Fig. 5. Its arrangement in the casing is shown in Fig. 6. The stationary nose is supported by three streamline struts. The stationary streamline continuation of the fan hub is centered in the casing by a spider of five radial blades and contains the ball-bearings which support the shaft at the fan end. The guide vanes are located between the fan and the radial blades. The guide vanes were designed after the fan was built and are based on measurements of air-discharge angles from the fan when operating at 3600 rpm and at capacities of 3000 and 4000 cfm. The angles were determined by the use of a directional tube,[8] and also by threads, and were measured at the three design sections. The measured angles are listed in Table 1. The inside and outside section readings are affected by boundary conditions. Guide vanes were made with angles corresponding to those measured at 4000 cfm and are shown in Fig. 7.

Test Results

The arrangements for the fan tests were substantially the same as those described in the paper by Marks and Weske[3] and illustrated in Fig. 6 of that paper. The only notable change is that calibrated orifices were substituted for the nozzle for measuring air and the fan shaft was halved in length, which made it possible to operate with speeds up to 3600 rpm.

Complete tests were made at 3600 rpm both with and without guide vanes and at 2400 rpm with guide vanes. Fig. 8 gives a comparison between the performance with and without guide vanes, while Fig. 9 gives a comparison between the performance at the two speeds.

The horsepowers shown in Figs. 8 and 9 and used in the calculations of efficiency are net horsepowers obtained by subtracting the friction horsepower from the dynamometer horsepower. Friction horsepower was determined by replacing the fan with a bladeless hub of the same form and weight

TABLE 1 MEASURED AIR-DISCHARGE ANGLES AT THREE DESIGN SECTIONS

Radius, in.	Discharge angle, deg 3000 cfm	Discharge angle, deg 4000 cfm
9.65	58	40
8.21	52	35
6.76	57	38

and observing the dynamometer horsepower required to rotate it at various speeds. When operating at 2400 rpm, the horsepower input is low and the friction horsepower is about 20 per cent of the dynamometer horsepower. Any considerable error in determining the friction horsepower would have an appreciable effect on the calculated value of the net horsepower at this speed. At 3600 rpm this source of possible error is negligible, since the friction horsepower is then a small fraction of the net horsepower. The question of possible error of the horsepower determinations at 2400 rpm is raised here because the peak efficiency at this speed is 82.5 per cent as compared with 77.5 per cent at 3600 rpm. This same variation of peak efficiency with speed was found in the three-bladed fan[3] and was commented upon as having been previously observed in the operation of propeller and centrifugal pumps.

[8] "The Determination of the Direction and Velocity of Flow of Fluids," by L. S. Marks, *Journal of the Franklin Institute*, vol. 217, 1934, p. 207.

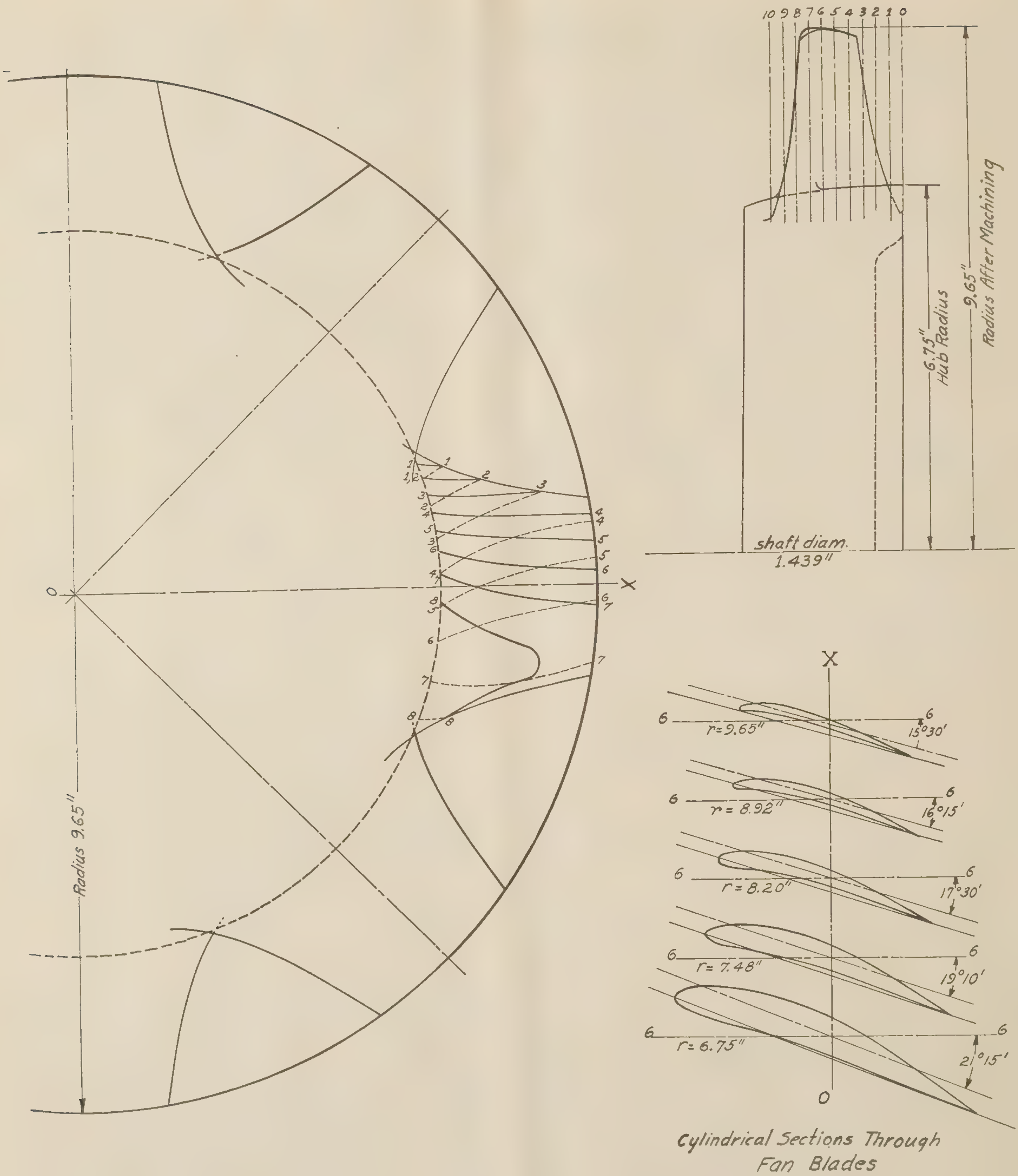

Fig. 5 Pattern Drawing of the Axial-Flow Fan Tested

The following observations may be made on the test results:

1 The static pressure at 3600 rpm and at peak efficiency is 4.7 in. of water, which corresponds to a static pressure of 22 in. of water with a tip speed of 40,000 fpm. This is an increase of 22 per cent over the static pressure obtained in the three-bladed fan.[3]

2 The net horsepower at no-flow is 16 per cent greater than the net horsepower at peak efficiency and compares with an increase of 125 per cent for the three-bladed fan.[3] This is a most important result since it permits the use of driving equipment of only half the power of that necessary for the three-bladed fan.[3]

3 The peak efficiency (total efficiency) is 77.5 per cent as compared with 80 per cent for the three-bladed fan.[3] However, a

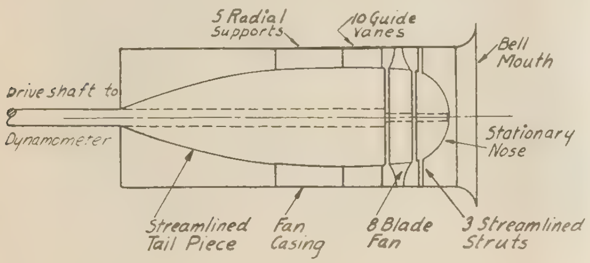

FIG. 6 FAN ASSEMBLY WITH STATIONARY NOSE AND TAIL PIECE IN ITS CASING

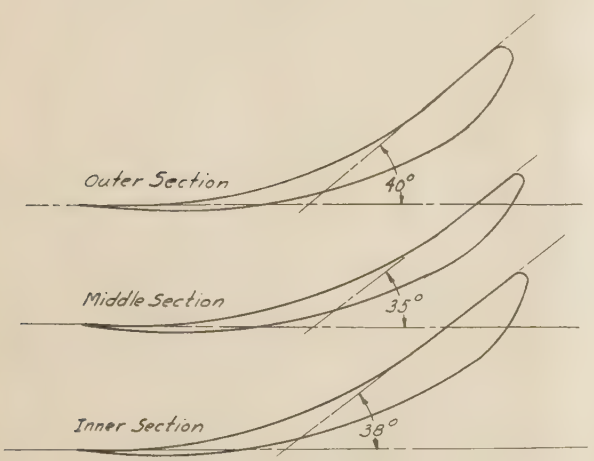

FIG. 7 GUIDE-VANE SECTIONS

comparison of peak efficiencies alone may be misleading; the efficiency is maintained better with the fan described in this paper than with the three-bladed fan.[3] For example, at half the capacity giving peak efficiency, the present fan has an efficiency of 54.5 per cent as compared with 45 per cent for the three-bladed fan. At capacities above that giving peak efficiency, the efficiency falls off very rapidly.

DISCUSSION OF TEST RESULTS

It is interesting to make a comparison of the test results with those predictable from the cascade-test data given in Fig. 1. As a sample procedure utilizing these data, take the outside section with a diameter of 19.3 in.; a N.A.C.A. 6306 blade section as shown in Fig. 4; a volume of 4000 cfm; an entrance velocity V_f of 65 fps; a speed of 3600 rpm; and a section tangential velocity U of 303 fps.

Assume $H_d = 520$ ft of air.

From Equation [4], $V_{u3} = 55.4$ fps.

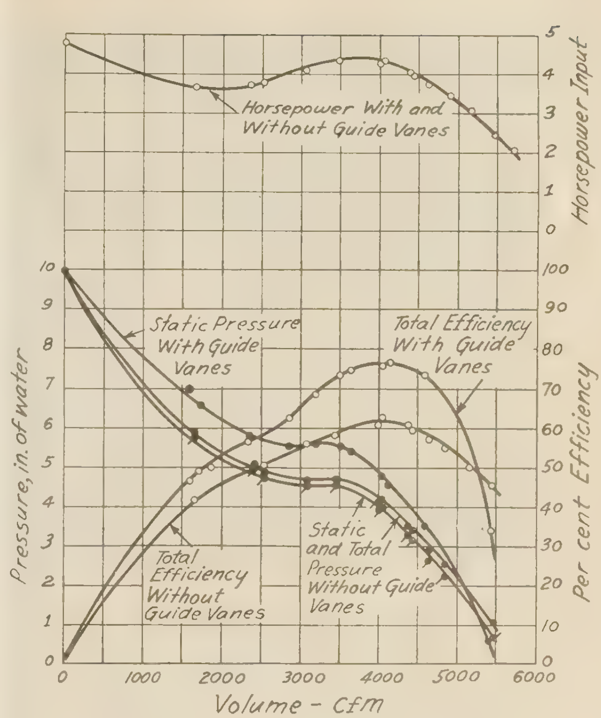

FIG. 8 PERFORMANCE CURVES OF AXIAL-FLOW FAN WITH AND WITHOUT GUIDE VANES

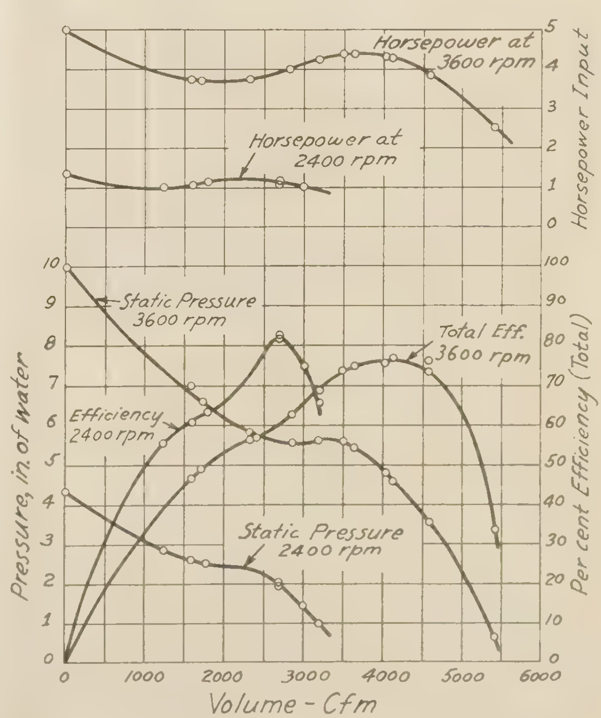

FIG. 9 PERFORMANCE CURVES OF AXIAL-FLOW FAN, WITH GUIDE VANES, AT 2400 AND 3600 RPM

From Equation [5], β = 13 deg 16 min, and since the blade pitch angle $(\beta + \alpha)$ = 15 deg 30 min as seen from Fig. 5, α = 2 deg 14 min.

From Fig. 4, $C_L = 0.775$ and $C_D = 0.0093$.

Interpolating in Fig. 1, the correction factor k for mutual blade interference is 0.610. The correction factor for C_D is negligible.

The lift coefficient normal to the long axis of the blade section

$$C_n = (kC_L \cos \alpha) + C_D \sin \alpha = 0.472.$$

The lift coefficient parallel with the axis of the fan

$$C_a = C_n[\cos (\beta + \alpha)] - C_t [\sin (\beta + \alpha)]$$

where $C_t = C_L \sin \alpha - C_D \cos \alpha = 0.455$

The velocity of the air stream relative to the blade section

$$V = \sqrt{[V_f^2 + (U - \tfrac{1}{2} V_{u3})^2]} = \sqrt{79{,}900} \text{ fps}$$

The thrust per blade $T = C_a \times V^2 \times d \times$ area of the blade lying between two cylindrical sections bordering closely the design section, where d is the density of the working fluid.

Multiplying T by the number of blades and distributing it over the area enclosed by the two bordering cylindrical sections

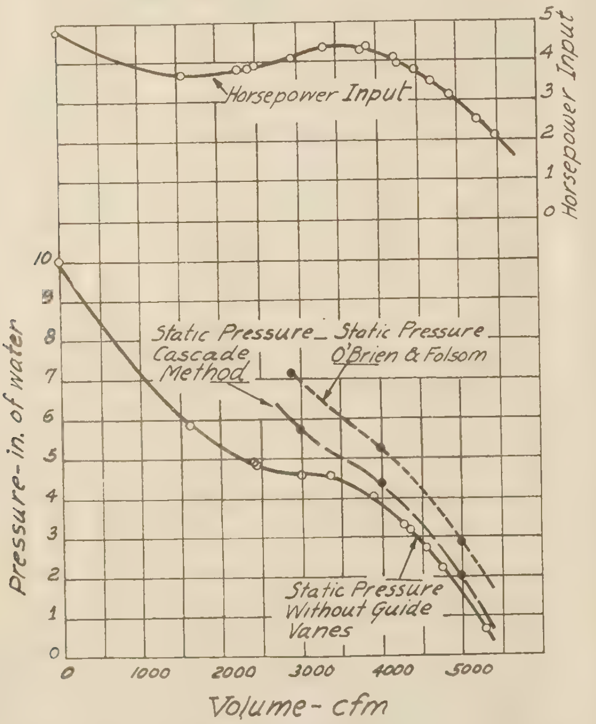

Fig. 10 Static Pressures Obtained in a Fan Without Guide Vanes Compared With Those Predictable by Two Design Methods

gives the developed head H_d, which for this particular case is found, after two more trial assumptions for H_d, to be 6.4 in. of water. The static head H_s is determined from Equation [2] and a weighted average is then obtained.

The static pressures as obtained by this procedure, together with those which were predicted from the O'Brien and Folsom formulations,[5] are shown in Fig. 10, the experimentally determined static pressures being given by the solid line. Both calculated curves show a strong similarity to the test curve but neither of them follows the shape of the test curve in the range of pressures above the point of inflection of that curve. For static pressures below the inflection point, the cascade method gives results which come very close to those of the test curve and the difference may well be ascribed to factors such as clearance and leakage losses, for which no corrections have been made.

It is interesting to note that the inflection in the test curve comes at a volume for which the angle of attack for all the blade sections is considerably below that at which break-off occurs,

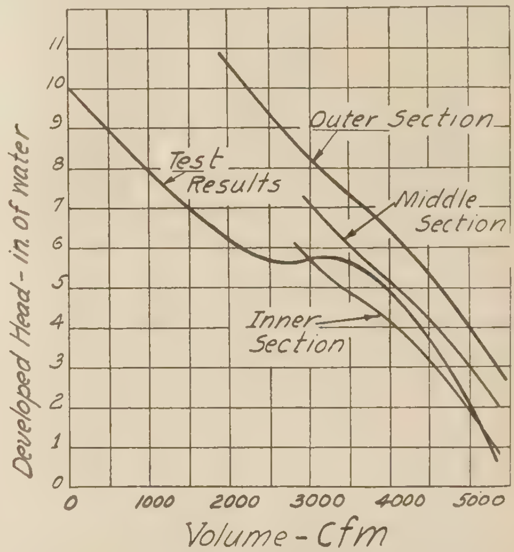

Fig. 11 Developed Head Obtained From Test Compared With Theoretical Values at Three Blade Sections

and is at a point where the horsepower curve is falling. This would seem to preclude the interpretation of this phenomenon as the breakdown point of the various blade sections.

In Fig. 11 are given the values of the developed head H_d at the three design sections as calculated from the cascade-test data. The test results for total head are also shown. The calculated developed heads do not have the same value at the three design sections under any condition of operation. The sections were designed by the procedure given by O'Brien and Folsom[5] in order to have a common value of developed head at 2900 cfm. At the peak efficiency, the developed head is seen to be 6.4 in. of water and falls to 4.2 in. at the hub. Such a pressure difference would result in recirculation between the blades, and therefore indicates one method of improving the efficiency.

The air-discharge angles at the middle section of the blades, as calculated from the cascade tests, are in very close agreement with the measured angles, being better than 0.25 deg. As calculated by the O'Brien and Folsom method,[5] they are about 8 deg 30 min too large. Near the hub and the tip a comparison is useless, as the boundary effect is unknown.

The noise intensity of this fan was not measured, but it was comparatively low when operated at capacities down to the inflection point of the static-pressure curve (2900 cfm at 3600 rpm). Below this point a penetrating wailing noise developed suddenly and persisted down to the no-flow condition.

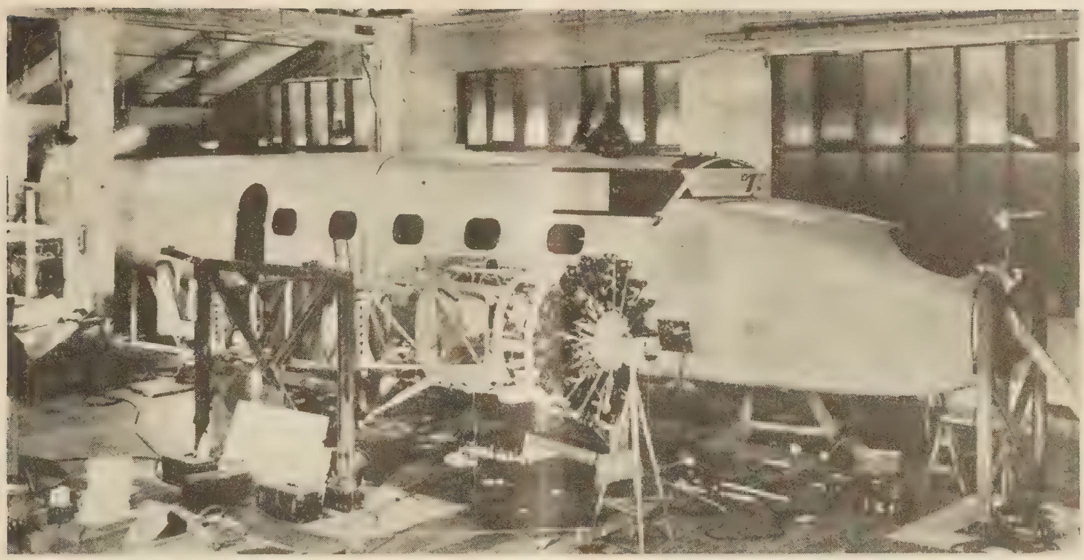

Fig. 1 Arc-Welded Engine Nacelle on Boeing Transport Plane

Reliability of Aircraft Welds

By N. F. WARD,[1] BERKELEY, CALIF.

This paper outlines the effects of fusion welding upon the dependability of structures in aircraft, due consideration being given to the existing conditions of welding practice found in this field. Obviously, the fusion welding of steel is of primary importance, since this is accepted practice. For the purposes of analysis, specimens prepared by experienced aircraft welders were subjected to physical tests and microscopic examination. With this information certain guiding principles in design for welding are outlined.

Introduction

Adequately designed airplanes, when fabricated by processes which produce safe carriers requiring a minimum of maintenance, represent the primary and substantial contribution to air transportation. The design of modern aircraft is guided largely by service requirements and economical fabrication processes which are consistent with structural stability.

The modern airplane is no simpler in structural form than its predecessor the tubular fuselage. In fact, the great variety of joints and fittings built in the all-metal airplane with prestressed covering, imposes additional fabrication costs which were absent in the welded airplanes of former years. Fusion welding, although contributing a reliable structural unit with favorable physical properties and the minimum of excess weight, is a fabrication process which is largely displaced by other methods with the advent of modern all-metal construction. The principal reason for the lessened emphasis on fusion welding is the vitiating effects of welding heats upon the alloys of thin section with further difficulties, in many instances, where low-melting-temperature alloys are required. Undoubtedly this field for structural application is inept for fusion welding, but entirely within the field of resistance welding. The successful use of resistance welding of thin sections in stainless-steel structures has been demonstrated by performance. Further extension of this type of welding to other metals is to be expected. However, fusion welding is utilized effectively from a structural and economical standpoint in the modern all-metal transport plane for such structural units as engine mountings, nacelles, shown in Fig. 1, fittings, and the landing gear.

Due to the complexity of structural units such as shown in Fig. 2, it is apparent that the usual fusion welding demands a precise skill during placement which only a manual operator can exercise. The degree of reliability in the weld is reduced to adequate training and experience of the welding personnel. At the present time, training is given at the aircraft manufacturing plants and only after successful welding of aircraft parts such as exhaust stacks, tanks, and fittings, is the experience recognized as sufficient to permit the operator to do structural welding. Through the efforts of the American Welding Society procedure

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Contributed by the Aeronautic Division and presented at the National Aeronautic-Hydraulic Meeting of The American Society of Mechanical Engineers at Berkeley, Calif., June 19, 20, and 21, 1934.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1935, for publication at a later date.

in aircraft welding has been revised and improved on the basis of experience, so that if closely adhered to, it should produce welds as reliable as human experience can devise.

Significance of Static Test

A study of failure under static loading such as encountered in the standard hardness and tension tests, although not simulated in actual service, serves with the aid of microscopic examination to detect inherent characteristics of welded joints. When these qualities in the weld and its immediate vicinity are recognized, suitable application of this type of joint is assured.

Fig. 2 Typical Gas-Welded Cluster Joint

For specific data in this connection, butt-welded tubes of cold-drawn chrome molybdenum S.A.E. 4130 steel were used. The tubes were 1 in. in diameter and 3 in. long, and had a wall thickness of 0.049 in. They were butt-welded with oxyacetylene torch and low-carbon-steel welding rod to form a specimen with an overall length of 6 ft. The welded tubes were fitted with steel plugs in both ends and inserted far enough to absorb the crushing load of the friction grips in the 30,000-lb Olsen Universal testing machine. The results of the commercial tests of the tubes, summarized in Table 1, are indicative of the reliability in static tension of the usual butt-type weld.

TABLE 1 TENSION TESTS OF BUTT WELDS IN 1-IN. TUBES

	Ultimate stress, lb per sq in.	Ductility in 2 in., %	Remarks
Minimum........	87,800	9.4	Broke in weld
Maximum.......	93,300	9.4	Broke in annealed section, see Fig. 3
Avg of six........	91,200	9.4	....

TABLE 2 COMPARISON OF HEAT-TREATED SPECIMENS OF BUTT-WELDED AND UNWELDED TUBES

	Ultimate stress, lb per sq in.	Ductility in 2 in. %	Remarks
Minimum..........	129,000	6.25	Broke in weld
Maximum..........	153,300	9.4	Broke in tube
Avg of six..........	142,400	7.3	....
Original tube as received.........	156,000	25.5	Broke at mid-section

After subjecting another group of butt-welded specimens to a combination of normalizing treatment at 1700 F hardening in water from 1700 F and tempering at 900 F, the results shown in Table 2 were obtained under static tension.

Prior to the tension tests and after the tubes were welded, a survey of hardness with a Rockwell hardness tester was made. The average values of three readings in the weld and at $^1/_4$-in. intervals either side of the weld were observed as indicated in Fig. 3. After heat-treatment the butt-weld hardness increased to Rockwell B-82 and the hardness either side of the weld remained uniform at average values of C-6. These tests indicate a softening or annealing in the region of the outer envelope of the welding flame which produces sufficient heat to recrystallize the cold-drawn microstructure. In the heat-treated tubes the variation in hardness was adjusted with the exception of the weld, where an increase in hardness was evident.

For static tests the indentation of the hardness penetrator appears to have little or no effect on the tube strength. Yet for the dynamical loading, encountered during fatigue tests, the indentation inhibits a weakness not already in the tube as welded. For this reason it was thought inadvisable to determine the hardness on fatigue specimens. Since the fatigue tests alter the physical structure of the metal, hardness determination after the test would be of little value. It remains for the microscopic examination to explain the allocation of the fracture as a result of the fatigue tests.

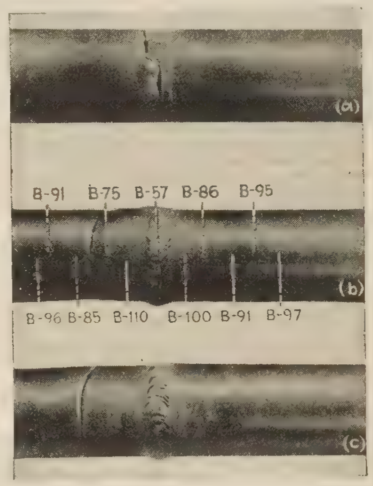

Fig. 3 Fractures in Tensile Specimen. Hardness Numbers at $^1/_4$-In. Intervals Before Tensile Tests Are Shown on the Center Specimen

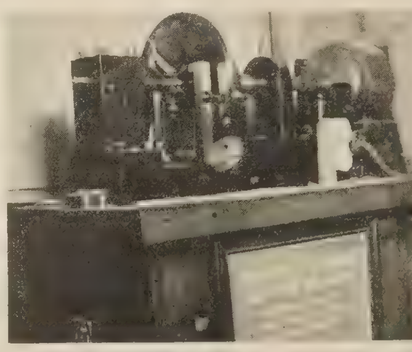

Fig. 4 Fatigue-Testing Machine for Tubes Under Bending Stress

Fatigue Tests of Tubes

With usual service conditions inducing repeated stresses, the endurance of welded tubes under repeated loading is a logical test. Due to limitations of grip size on the fatigue-testing machine which was available, it was necessary to use $^1/_2$-in. chrome-molybdenum S.A.E. 4130 tubes with 0.049-in. wall thickness. Since the smaller tubes are more difficult to weld accurately, the results should represent less favorable performance than the average. In other words, the minimum values for design are indicated by these tests.

The fatigue machine shown in Fig. 4 is of the cantilever type in which one end of the specimen is held rigidly while the other end is guided and loaded with an 80-lb indicator spring. The spring load deflects the specimen and induces the maximum stress at the inner side of a rigid clamp. In the tube tests, the maximum stress is in the region affected by the welding heats. The mounting of the tubular specimen requires the steel plug on the clamped end to prevent collapse of the tube under the grip pressure. A similar plug is inserted in the opposite end which engages the ball-bearing housing through which the spring load is applied.

Each heat-treated specimen, as welded with oxyacetylene flame, was tested for a given spring deflection which was measured on the dial gage shown in Fig. 4. By means of a direct-connected $^1/_4$-hp motor, the single specimens were rotated at 1725 rpm until fractured. This was possible by subjecting each specimen to a different stress by proper spring adjustment. The results are plotted on the curves in Fig. 6. For maximum

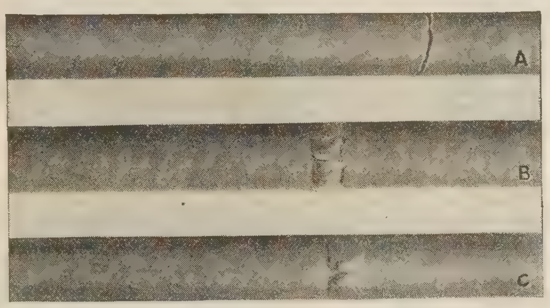

Fig. 5 Fatigue Failure of $^1/_2$-In. Cr-Mo Steel Tubing
(a, tube as received; b and c, Oxyacetylene-welded tubes.)

deflection the specimens fractured as low as 62,000 reversals. For stresses of low intensity some of the specimens did not break after 3,000,000 reversals. In every instance the specimens as welded broke adjacent to the weld bead as shown in Fig. 5. On which side of the weld bead failure would occur wasn ot predictable. Some failed on the side nearest to the rigid grip and an equal number of other specimens failed on the opposite side of the weld bead, as shown by comparing Fig. 5c with Fig. 5b. The fracture in the unwelded tube may be seen in Fig. 5a.

Several observations for butt-welded tubes during these comparative tests show that loads inducing high stress are accompanied by failure during a wide variation of cyclic frequency.

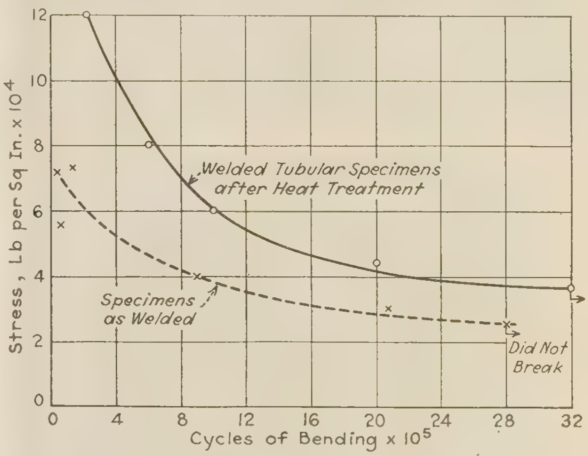

Fig. 6 Fatigue-Test Curves of $^1/_2$-In. Cr-Mo Oxyacetylene-Welded Tubes With 0.049-In. Wall Thickness

The heat-treated specimens develop a more uniform response to fatigue in bending. Improvement of the endurance limit by heat-treatment of welded tubes in these tests amounted to approximately 40 per cent. The ratio of endurance strength to static tensile strength of the tubes as welded was $^1/_4$, while for welded tubes after heat-treatment the ratio was $^1/_3$.

Within probable errors of the test, this comparison between the tubes as welded and the welded tubes after heat-treatment maintains a close parity for the ratio of endurance limit to ultimate tensile strength. However, the fatigue failure in the butt-welded tube without subsequent heat-treatment has a tendency to localize failure in the immediate vicinity of the weld as indicated by the fractures in Fig. 5. The fatigue failure in the heat-treated tubes appeared most frequently at least the diameter of the tube from the weld. Obviously, the desire in service is to preserve these critical sections against concentration of load by any of several types of construction, such as gusseting or reinforcing with straps or telescoping the tubes or using a fishmouth joint.

Where these welds fail has been demonstrated by physical tests. Why the welds are more reliable in certain sections and vitiated in others is revealed in the microstructure of the weld and its vicinity.

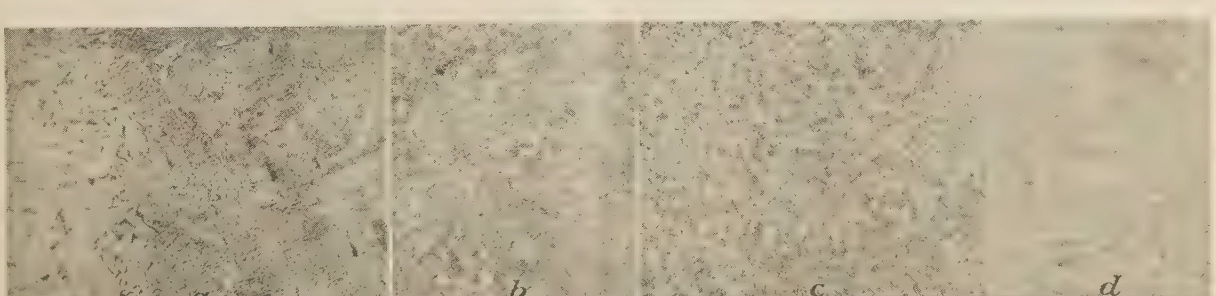

Fig. 7 Microstructure in Oxyacetylene-Weld (× 130)
(a, weld matrix at left with transition beginning at the right; b, transition; c, annealed section $^1/_2$ in. and $^3/_4$ in. from weld; d, original cold-drawn tube

Influence of Welding Heats

Progressing from the weld outward toward the parent metal, the metallurgical changes in the weld may be seen in Fig. 7. The large areas of free ferrite are present with less carbon matrix than seen in the original tube in Fig. 7*d*. The weld metal is a typical structure found in low-carbon-steel castings cooled slowly from fusion temperatures. The precipitation of large ferritic masses accounts for the soft weld. The heat used in placing the weld has been moderate, since no excess of oxide and nitride contamination is evident. The avoidance of high temperature in placing the weld is essential to keep the gas (oxygen and nitrogen from the air) content at a minimum. These gases develop a porosity and form oxides and nitrides which embrittle the weld if present in combination.

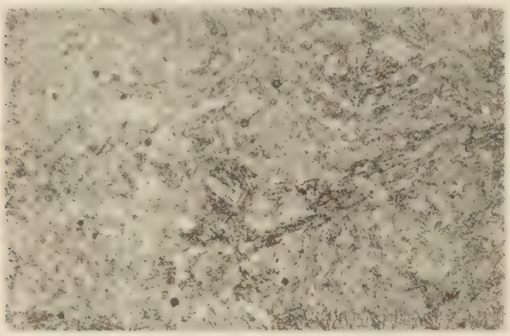

Fig. 8 Microstructure in Transition Zone of Heat-Treated Weld

The transition zone from weld to tube appears in Figs. 7*a* and 7*b*. The microstructure is large grained and of tough consistency which is harder than the weld except at grain boundaries, which are soft ferrite. The growth of large-grain structure is a natural accompaniment of welding heats. The precipitation of tough constituents in this transition zone is largely due to rapid conductivity of heat from the weld to the cooler tube. The cross-section of a weld thus affected with the aircraft welding torches persists for $^1/_2$ in. either side of the welds in varying intensity, which depends upon the adjustment of the torch flame and its position. The larger grain of the weld in comparison with the transition zone represents a source of weakness with reversal of loading, since it yields more rapidly than the hard tough metal adjoining it in the transition zone.

Interposed for a distance between $^1/_2$ in. and $^3/_4$ in. between the transition zone Fig. 7*a*, *b* and the original plate 7*d* is found the recrystallized section Fig. 7*c* in which refinement of the grain has been produced by contact of a portion of the welding flame or the heat as conducted from the weld to the cooler plate or from both causes. The existence of this structure explains the fracture in this area for sound weld during static-tension tests, rather than in the weld. Briefly, the unit stress is least in the weld because of its greater section. The softened section (Figs. 3 and 7*c*) strain-hardens with the application of load but suffers a greater reduction in cross-sectional area than adjoining sections, thus developing greater stress intensity which persists under load until the ultimate stress is reached and pronounced reduction in cross-sectional area continues until fracture occurs.

Since the base or parent metals in aircraft structural members are cold-drawn or rolled, they reach their yield point at a greater load than for the annealed section. Where there is a weld fracture under static-tension load, the specimen has been misaligned or the weld has been overheated and consequently embrittled by contamination of oxides and "cold shuts."

Summary of Physical Changes in Oxyacetylene Weld

In tubing of chrome-molybdenum steel the welding heat is beneficial to certain sections and detrimental to others. Since chrome-molybdenum steel hardens as it cools in air from fusion temperatures, the static strength of the areas adjacent to the weld approaches or exceeds that of the cold-drawn sections. Use of mild-steel tubing for welded joints effectively prevents as wide a variation in hardness, but the strength averages half that of chrome-molybdenum steel. The fatigue resistance of the fused areas is reduced because of the large grain. Interposed between this zone and the cold-drawn section is an annealed section in which sound welds finally fracture under static loading.

Welded-tube sections which undergo proper heat-treatment, i.e., normalizing hardening and drawing, develop high ultimate strength, as is evident from the values in Table 2. This heat-treatment develops a welded structure as reliable as the original by adjusting inequalities of the crystalline matrix and producing uniform strength and ductility. The microstructure changes at the weld zone resulting from this heat-treatment are shown in Fig. 8. The position and type of fracture for the static-tension tests is shown for selected specimens in Fig. 9.

Fig. 9 Typical Failure of Oxyacetylene-Welded Tubes After Heat-Treatment

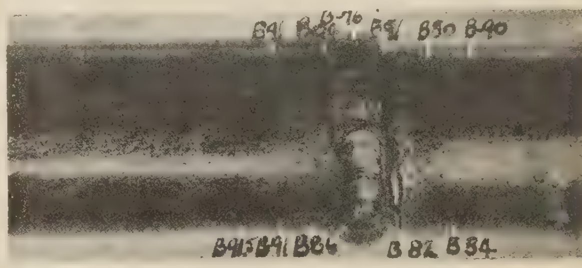

Fig. 10 Static-Tension Failure in Arc-Welded Tube. Hardness Values Taken at $^1/_8$-In. Intervals

Characteristics of Arc Welds

Metallic arc welding with direct current and low-carbon-steel electrode has had limited application in airplane structures. For adequate fusion of the tube or plate the bead deposit is greater than for oxyacetylene welding. The concentrated heat evolved from the arc localizes the critical zone in close proximity to the weld beads. The annealed section, which is less extensive than that for the oxyacetylene welds, develops within $^1/_4$ in. of the welds. The variation in hardness appears to be less marked

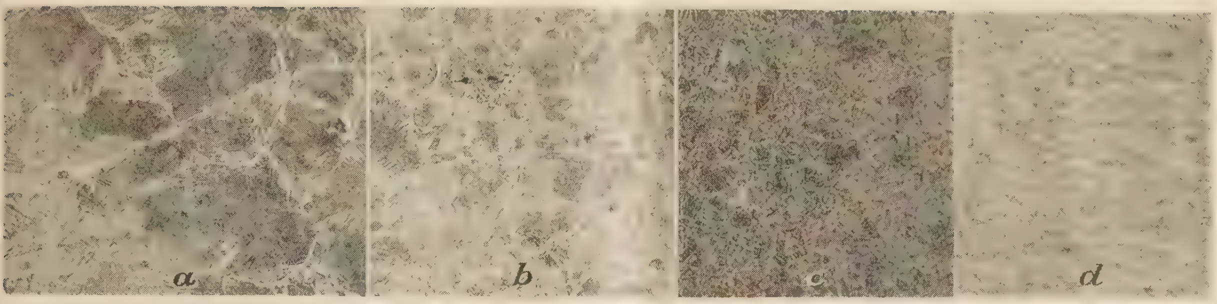

FIG. 11 TYPICAL MICROSTRUCTURE IN 1-IN. TUBE ARC WELDED WITH DIRECT CURRENT AND LOW-CARBON FILLER ROD (× 130)
(*a*, weld matrix showing on the right large-grained sorbite with ferrite boundaries; *b*, transition zone; *c*, partially annealed zone 1/4-in. from weld; *d*, original cold-drawn tube.)

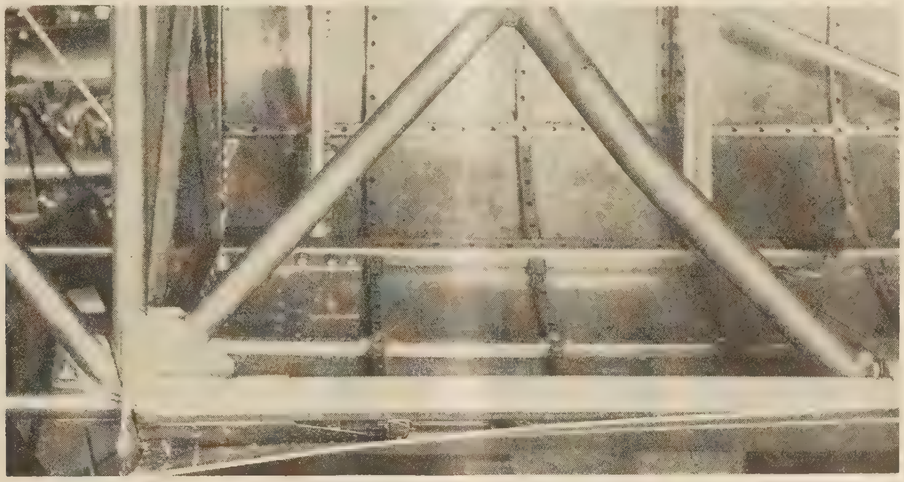

FIG. 12 STATIC FAILURE IN BEADING OF ARC-WELDED TUBE. LOAD FACTOR = 14 1/2

than for the oxyacetylene-welded tube. Comparison of Rockwell hardness values for the electric arc weld is indicated at 1/8-in. intervals in Fig. 10. The failure under static-tension loading is evident from Fig. 10.

The transformations in the weld and tube for a representative arc-welded specimen are shown in Fig. 11. The weld structure at the left Fig. 11*a* is a typical steel matrix as cast, tapering into a sorbitic zone of large grain bounded by soft ferrite. The retention of high sorbitic content is probably due to the large temperature gradient between the weld and its environs which produces faster chilling than is present in oxyacetylene welding. There is evidence in Fig. 11 that grain diminishes in size until the annealing temperature is reached about 1/4 in. from the fusion zone. Then the original cold-rolled structure is resumed with lowering of the temperature below that required for recrystallization. Fracture in static tension for the butt-welded tubes occurs in this annealed zone adjacent to the weld for reasons given under the discussion for oxyacetylene welding. The zones of fusion and transformation are very similar for both the oxyacetylene and direct-current arc welds.

The frequency with which failures occur outside of the weld when tested with the weld bead intact for a joint properly designed and welded, is evidence that the strength is not dependent upon the welds so much as the amount of weakening due to the welding process.

OTHER FACTORS AFFECTING RELIABILITY OF WELDED JOINTS

Temperature Stresses. There are many conditions arising from mechanical shrinkage as the welds cool. A brief for the solution of the problems encountered could not be attempted in this short treatise. The principal difficulty with shrinkage in welds during solidification is the adjustment of the residual stresses during cooling or immediately after the weld is placed. Usually the intensity of stress due to shrinkage is localized in the sections which cool unevenly. For example, the hotter areas are weak and the stress is transferred from the cool, stronger areas. Where the correct sequence of deposition of the weld bead and tack welding is used, the shrinkage is absorbed sufficiently so that subsequent normalizing, stress-relief annealing or peening adjusts the residual stresses in larger areas. In this way the stress intensity is distributed but never eliminated. Abrupt change of sections, such as heavy reinforcement in the weld bead, is undesirable not only because of the diminished fatigue resistance at the juncture of the bead and the tube, but also because of localization of shrinkage in this larger mass which retains its temperature during a longer period than the thin tube. Weld beads which are flush with the tube and rough or undercut are prone to weakness in fatigue and represent doubtful security in a structure.

Significance of Welding Knowledge for Design. The general conclusions which may be drawn from the foregoing facts about fusion welding in aircraft leads to the proportioning of the structural members so that stresses of a concentrated nature are not imposed at the critical section. By the critical section is meant that at which failure originates due to predisposition to structural weakness under repeated tension, compression, shear loading, or their combination. Clarifying examples may be cited.

The structural member or members in a fuselage are under tension, compression, and bending loads of varying intensity. The tube size, or plate thickness, is calculated for one or several predominant stresses. In a typical instance of a tube where a large ratio of length to diameter is used, the welds at the ends can safely develop the tube strength, unless the joints receive heat treatment; otherwise the crushing strength in the annealed section of the weld is reduced below that of the tube. This critical section requires suitable reinforcement, such as a strap or gusset. With adequate weld reinforcement the tube should bend in the center. Since the stress is seldom equally distributed in the final welded unit the probable failure occurs near the center section which, if very critical, transfers intensity to the tube ends. The static test on a fuselage shown in Fig. 12 indicates a collapse of the tube under a load factor of $14^1/_2$ with the end welds remaining intact.

Further extension of the analysis for the welded construction explains why application of heat unnecessarily near critical sections should be avoided. For instance, holes drilled for injecting lionoil to prevent internal corrosion of structural tubes are located usually beyond the first critical section and the drill holes are filled with screw plugs to avoid softening by flame or arc heat which produces areas susceptible to collapse near the critical section. Welding of telescoped tubes or fishmouth is preferred to butt-welded tubes because of the added stiffness and stability afforded in the joint. Deposition of relatively smooth concave weld beads is done where vibration is a predominant consideration. These show the place of accurate knowledge of weld performance as briefly outlined in the preceding discussion.

Acknowledgments

The author wishes to take this opportunity to express his appreciation to Mr. Retan of the Boeing School, for his assistance in preparing several of the representative weld specimens; to the Boeing School of Aeronautics, for the supply of tubing used in the specimens; and to the Department of Engineering Materials of the University of California, for the use of physical testing equipment.

The Viscosity of Water and Superheated Steam[1]

By G. A. HAWKINS,[2] H. L. SOLBERG,[3] and A. A. POTTER,[4] LAFAYETTE, IND.

The authors present in this paper data on the viscosity of water and superheated steam at temperatures and pressures used in modern power-plant practice. The authors constructed an apparatus to measure the viscosity of water and superheated steam at pressures up to 3500 lb per sq in. abs and at temperatures up to the critical in the case of water and between 800 F and 1000 F for superheated steam. The apparatus is described and the results obtained with it are reported in this paper.

THE increasing use of dimensional analysis in correlating data and solving problems in the fields of heat transfer and fluid friction has made viscosity a very important physical property of fluids. In making a survey of the literature preparatory to an investigation of flow distribution in forced-circulation once-through steam generators (1),[5] the authors found that data on the viscosity of water and superheated steam are very meager or non-existent at the temperatures and pressures which are used in modern power-plant practice. The viscosity of water is known quite accurately at temperatures below 212 F and some determinations have been made at temperatures up to 320 F. von Hevesy (2) calculated the viscosity of water at elevated temperatures on the assumption that viscosity is a function of electrical conductivity, but this assumption has not been verified experimentally. M. DeHaas (3) used a capillary-tube viscometer for determining the coefficient of viscosity of water up to 153 C. The viscosity of superheated steam at atmospheric pressure is known with reasonable accuracy. Speyerer (4) has determined the effect of pressure on the viscosity of superheated steam at pressures up to a maximum of ten atmospheres.

In view of the absence of experimental data beyond the above limits, the authors decided to construct a suitable apparatus to measure the viscosity of water and superheated steam at pressures up to 3500 lb per sq in. abs and at temperatures up to the critical in the case of water and between 800 F and 1000 F for superheated steam. The results are reported in this paper.

Since this investigation was started Schiller (5) has published viscosity determinations. Schiller obtained the relation for calculating the viscosity of steam in terms of that of water, by observing the velocity at which the discharge coefficient of a nozzle shows an abrupt increase for water and steam and then equating the two Reynolds numbers. Schiller's curves show one test point at 30 atmospheres pressure, four test points at 25 atmospheres and other points at lower pressures with a maximum temperature of about 540 F.

Selection of Type of Viscometer[6]

In deciding upon the type of viscometer to be used in this investigation, where pressures of 3500 lb per sq in. and temperatures of 1000 F were contemplated, the matter of safety was given first consideration because of the high energy storage per unit of volume of hot compressed water. The apparatus had to be suitable for a wide range of viscosities and reasonable in cost. Adaptability to the use of automatic recording devices was desirable in order to reduce the human element in the collection of the more essential data. The high pressures involved, the effect of high temperatures on the elastic properties of suspensions, difficulties in measuring small pressure differences accurately, and of maintaining definite surfaces of separation between liquids and condensable vapors, resulted in discarding most of the types of viscometers which have been used for viscosity determinations.

The Lawaczeck (6) viscometer was selected as being the most suitable type for the range of conditions encountered in these tests. This viscometer was developed in 1919 and consists essentially of a metal weight falling in a tube closed at its lower end and having a diameter which is slightly greater than that of the weight. As the weight falls, the liquid is made to flow with streamline motion through the annular space between the weight and the tube. There are three distinct resistances to the fall of the weight through the liquid. The form of the streamlines causes a so-called "head resistance" which can be decreased to a minimum by using a long weight and a small clearance. As the liquid flows through the annular space, a pressure gradient is set up similar to the case of the flow of a fluid through an annular channel. This resistance is fixed by the area of the cross-section and the rate of fall of the weight. The third resistance is a viscous drag due to the relative movement of the two cylindrical walls. Neglecting the head resistance, the formula for this viscometer is

[1] Progress Report A.S.M.E. Special Research Committee on Critical-Pressure Steam Boilers.

[2] Assistant, Engineering Experiment Station, Purdue University. Jun. A.S.M.E. Mr. Hawkins received part of his college training at the Colorado School of Mines and the degrees of B.S. and M.S. in mechanical engineering and also his Ph.D degree from Purdue University.

[3] Associate-Professor of Mechanical Engineering, Purdue University. Mem. A.S.M.E. Mr. Solberg holds the degrees of B.S. and M.S. from Purdue University. He has been for the last 14 years on the teaching staff of Purdue University where in his present position he is in charge of the instruction in power-plant engineering and thermodynamics. He is author of numerous articles on power engineering in the technical press.

[4] Dean of Engineering and Director of Engineering Experiment Station, Purdue University. Past-President A.S.M.E. Dr. Potter is a graduate of Massachusetts Institute of Technology; he has had 32 years' experience teaching and practice; he was connected with the General Electric Company during the earlier development of the steam turbine. For 15 years he was Professor and Dean of Engineering at Kansas State College, and since 1920 has held his present position. He is author of books, articles, and papers on thermodynamics and heat engineering.

[5] Numbers in parentheses refer to similarly numbered references in the bibliography given at the end of this paper.

Contributed by the Power Division and Research Committee for presentation at the Annual Meeting of The American Society of Mechanical Engineers to be held in New York, N. Y., December 2 to 6, 1935.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

[6] The description of the development of the viscometer and the values for the viscosity of water are based on a thesis submitted by G. A. Hawkins in partial fulfillment of the requirements for the Ph.D. degree at Purdue University, June, 1935. The work on the viscosity of steam was completed subsequent to the submission of this thesis.

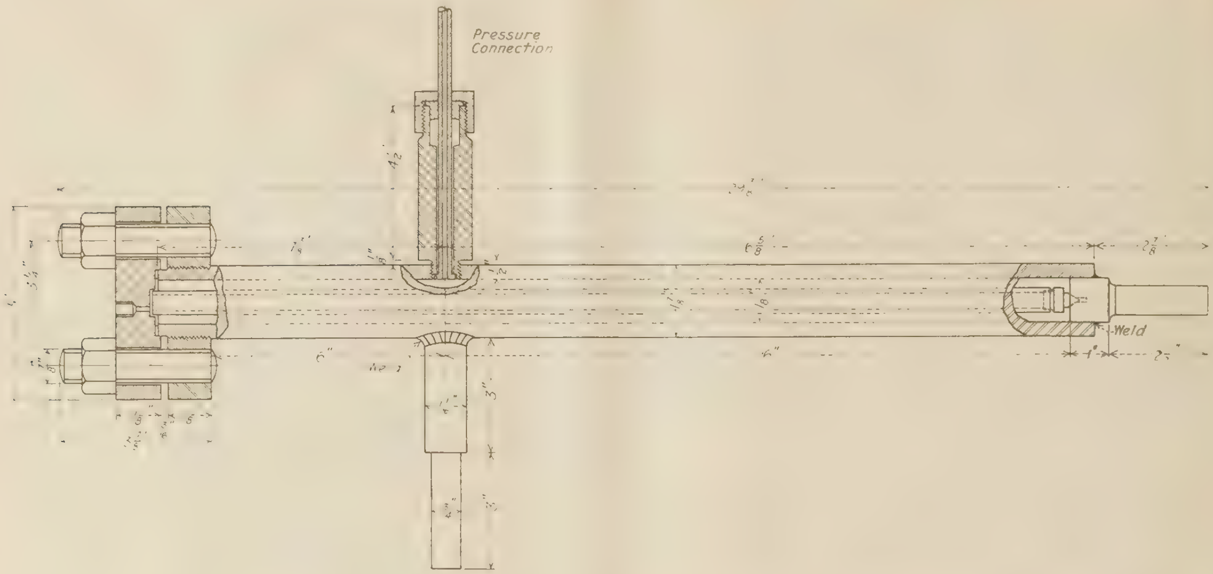

FIG. 1 DETAIL OF VISCOMETER AND PRESSURE CONNECTION

$$\mu = \frac{t}{s} (\sigma - \rho) \frac{\delta^3 d}{3(d + 2\delta)^2 - (2\delta)^2} \quad [1]$$

where μ is the absolute viscosity; σ is the density of the weight or fall-body; ρ is the density of the fluid; δ is the annular space between the fall-body and the tube; d is the diameter of the fall-body; s is the fixed distance through which the weight falls inside the tube; and t is the time required for the weight to fall the fixed distance s.

Because of the difficulty of measuring the dimensions accurately, the instrument is calibrated by using a fluid of known viscosity. Using a calibration constant C determined in this manner, Equation [1] is changed to the form

$$\mu = C (\sigma - \rho) t \quad [2]$$

Bridgman (7) in 1926 used a method similar to this for determining the viscosity of liquids at very high pressures but at low temperatures. In 1933, Stakelbeck (8) used a Lawaczeck viscometer for measuring the viscosity of carbon dioxide, ammonia, and sulphur dioxide, at the suggestion of R. Planck.

DESCRIPTION OF THE VISCOMETER

The viscometer, shown in Fig. 1, consists of an accurately bored, stainless-steel vertical tube through which a weight, or fall-body, of slightly smaller diameter is moved under the influence of gravity. The tube is placed concentrically within a seamless steel pipe capable of holding safely the maximum pressures to be used. This outer pipe or pressure vessel, hereafter called the container, was closed at the lower end by welding and was fitted with flanges at the upper end to allow for insertion of the tube and fall-bodies, and for periodic inspection. The top was closed with a blank flange and was provided with a $^1/_4$-in. vent valve. Two trunnions were secured to the container to serve as a support about which the apparatus could be rotated through an angle of 180 deg. One of the trunnions was provided with a packed pressure connection to which the pressure gage was attached. Distilled water was forced into the container by means of a hydraulic pump. The temperature of the fluid could be regulated by means of an electric heating coil which was wound around the container and thoroughly insulated. Fig. 2 shows the assembly

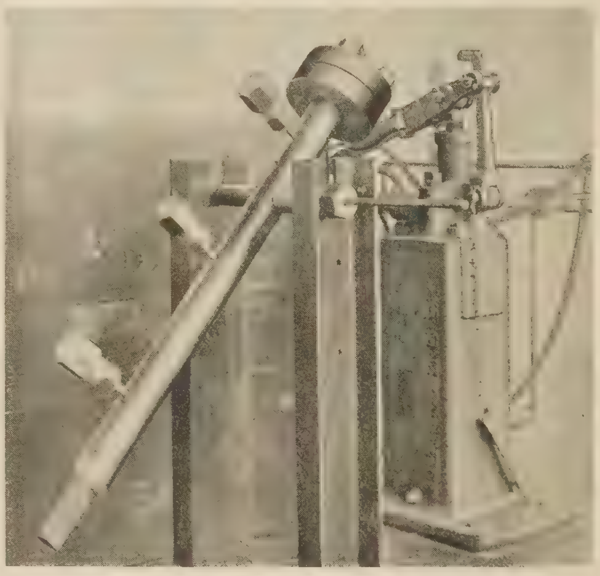

FIG. 2 VISCOMETER TEST ARRANGEMENT

of the container as mounted in its bearings. The three boxes for the electrical connections to the two timing coils are also shown. Three $^1/_8$-in. iron pipe taps were provided in the container, one at the center and one near each end, for the insertion of thermocouples.

The tube, 23 in. long and 0.658 in. outside diameter, was made by the Winchester Repeating Arms Company from a piece of Carpenter Steel Company No. 8 alloy containing 18 per cent chromium, 8 per cent nickel and a small percentage of selenium to facilitate machining. The bar was drilled, reamed four times and polished to 0.410 in. internal diameter with a high-speed lap coated with emery and oil. Periodic inspection of the tube has shown that it successfully resists corrosion and oxidation when in contact with steam and water at high temperatures. One end of the tube was threaded with a standard pipe tap and fitted

with a stainless-steel plug which centered it in a conical-shaped socket in the bottom of the container so as to insure coaxial alignment of the tube with the container when the two were assembled.

Details of the fall-bodies which were used in this investigation are shown in Fig. 3. They were drilled hollow to reduce their weight and the outside diameter is such that the Reynolds number of the fluid flowing into the annular space between the fall-body and the tube never exceeded 1300. The time required for a fall of 14 in. varied from 10 minutes to as low as 2 seconds, about 50 per cent of the tests involved a time of less than 1 minute. Fall-body No. 28 was used for determining the viscosity of water and this was checked by fall-bodies Nos. 30 and 31 in order to establish the fact that the results are independent of the shape of the ends of the fall-body. Fall-body No. 35 was used for determining the viscosity of superheated steam, which was checked by fall-body No. 33.

Pressures below 1500 lb per sq in. were measured by means of calibrated Bourdon gages. A Bailey self-loading, deadweight gage was used for higher pressures.

Temperatures were measured by iron-constantan thermocouples welded into the bottom of a steel well which was threaded to fit a $^1/_8$-in. pipe tap, as shown in Fig. 4. The thread on the thermocouple body was sufficiently long so that the hot junction was very close to the tube which contained the fall-body. Twenty thermocouples were made and calibrated in the physics laboratory of Purdue University against a platinum, platinum-rhodium thermocouple which had been checked by the National Bureau of Standards. Two of the three thermocouples were removed in rotating pairs at regular intervals and replaced by others, the third one being left in place as a check on the new one. Thermocouples were also checked periodically by boiling water in the viscometer with the vent valve open to the atmosphere. The thermocouples were connected to a very sensitive high-resistance Wilson-Maeulen millivoltmeter; the couple resistances were measured with a Wheatstone bridge.

Timing Equipment

The principal difficulty which was encountered in developing the viscometer was in devising an accurate means of timing the

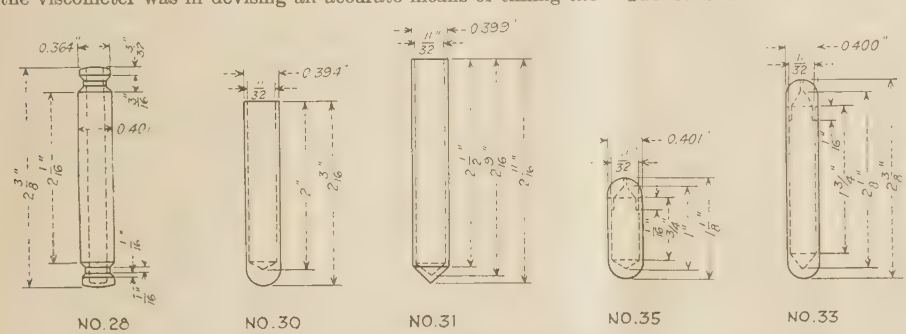

Fig. 3 Details of Fall-Bodies

rate of fall of the body which was dropping through the tube. Among the schemes which were investigated and abandoned may be mentioned the use of an X-ray and Geiger-Müller counter, an X-ray and sensitive screen, and an oscillator, receiver, and oscillograph. An a-c bridge amplifier and recording milliammeter were finally selected as the most satisfactory solution to the problem.

The a-c bridge circuit is shown in Fig. 5. The two coils, indicated in the circuit diagram, were wound on the tube through which the fall-body dropped and were spaced about 14 in. apart. When the fall-body passed through either coil, the circuit was unbalanced sufficiently to swing a pen across a strip chart on the recording milliammeter. Each coil consisted of a single layer of 60 turns of copper wire wound on a sheet of mica rigidly held in place on the tube by two steel clamps which also served as terminals for the ends of the coils. Each turn was separated from the adjacent turn by a small asbestos thread. After winding,

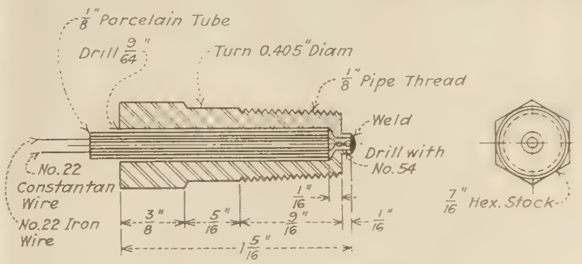

Fig. 4 Design of Thermocouples

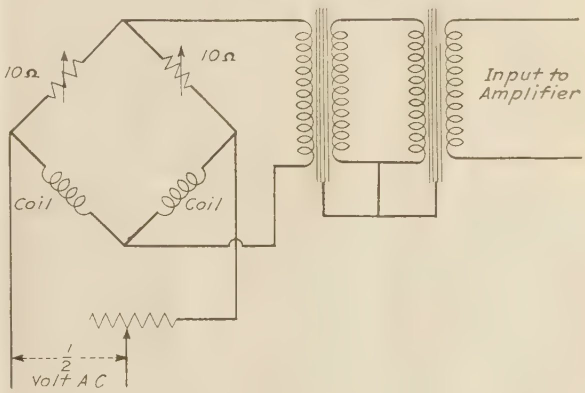

Fig. 5 A-C Bridge Circuit

the coils were examined for defects with a magnifying glass. They were as nearly identical as it is possible to make them. The coils were connected in series and the three leads were brought out in asbestos tubing through the pressure connections shown in Fig. 2. Two identical 10-ohm rheostats were used for balancing the bridge. An objectionable 60-cycle disturbance which was being picked up by the bridge, was eliminated by connecting the output of the bridge to two transformers, as shown in Fig. 5. A constant voltage had to be impressed on the bridge in order to obtain satisfactory results.

Fig. 6 illustrates the circuit of the amplifier. It consisted of two 57 amplifiers and one 45 triode power-amplifier tube. In order to eliminate the d-c part of the power output of the amplifier, the leads of the output were connected to the primary of a small transformer. The secondary output, consisting of the a-c part of the amplifier output, was rectified by a small Rectox rectifier. The rectifier output was connected through a double-pole double-throw switch to an indicating milliammeter having a range of 0 to 100 ma and to an Esterline-Angus recording milliammeter having a range of 0 to 5 ma.

The recording-meter chart of the Esterline-Angus milliammeter

was driven by a synchronous motor from the power lines of the Public Service Company of Indiana. System frequency control in the case of this public utility is obtained from Chicago and is excellent as shown by the records from recording frequency meters in the electrical-engineering meter laboratory of Purdue University. The effect of any momentary variations in frequency was averaged out by taking from 5 to 40 readings at each test point. Chart speeds of $^3/_4$, $1^1/_2$, 3, 6, and 12 in. per hr and per min were

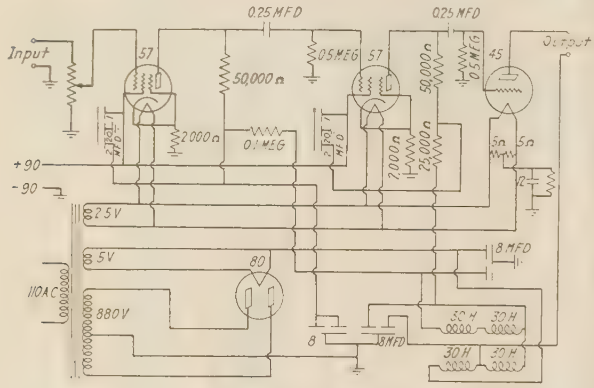

Fig. 6 Circuit Diagram of the Amplifier

obtainable from the recorder. Charts were driven at 12 in. per min except where the time of fall exceeded five minutes when lower chart speeds were used. With a chart speed of 12 in. per min, it was possible to scale the time to about one-tenth of a second. The chart speed was checked periodically by recording with a stopwatch the time between two interruptions of the bridge circuit and scaling the distance on the chart.

Fig. 7 shows a typical chart. The two humps are due to the unbalance created in the bridge circuit as the fall-body passed through the coils. The time of fall was scaled from each chart at five different points such as aa', bb', etc. in Fig. 7. If any of the five time measurements differed from the average by more than 1 per cent, the test was discarded.

Some difficulty was experienced at first in securing symmetrical curves from the two coils because of unequal heating of the viscometer. A variation in temperature of 2 F between the two coils altered their resistances sufficiently to unbalance the circuit and destroy the symmetry of the curves. Proper arrangement of heating coils on the container solved this problem. No tests were made unless the readings of all three thermocouples agreed within 1 deg.

A fall-body was suspended from a chemical balance in one of the coils while the input to the bridge circuit was varied in order to check the possibility of these coils disturbing the rate of fall of the body. No deflection of the balance was obtained until the voltage impressed on the bridge circuit was 30 times that used in the test.

The coils were mounted 14 in. apart on the tube and far enough from each end of the tube so that the fall-body had accelerated to a uniform velocity before it entered the tube section which was surrounded by the coil.

Calibration Tests

A comparison of Equations [1] and [2] shows that the term C in Equation [2] is a function of the dimensions of the fall-body, tube, and distance between timing coils. These dimensions cannot be measured with sufficient accuracy and are also subject to variation with temperature. The effect of compression of the steel due to pressure was neglected as the error so introduced was less than 0.02 per cent at 3500 lb per sq in. The effect of temperature may be treated as a linear function of the temperature without introducing an error of more than 0.5 per cent at 1000 F. The term C for each fall-body was determined by timing the fall when a fluid of known density and viscosity was placed in the viscometer and heated to various temperatures. The results were expressed by an equation of the form

$$C = a + bt$$

where a and b are constants and t is the temperature of the fluid in degrees fahrenheit. These calibration constants for the various fall-bodies are given in Table 1.

TABLE 1 CALIBRATION CONSTANTS OF FALL-BODIES — $C^a = (a + bt)$

Fall-body, No.	a	b
28	0.3628	0.000086
30	0.2525	0.000057
31	0.3560	0.000060
35	13.4200	0.019500
33	19.2700	0.016500

Note: t is in degrees F.

[a] See Equation [2].

Since fall-bodies Nos. 28, 30, and 31 were used to measure the viscosity of water, their constants were determined by using water at temperatures up to 212 F within which range the viscosity and density of water are known quite accurately. These values for viscosity were obtained from the International Critical Tables.[7] These calibration constants subsequently gave data on the viscosity of water which were identical for the three fall-bodies and which were in excellent agreement with the published

[7] International Critical Tables. Published for the National Research Council, McGraw Hill Book Company, New York, 1929, vol. 5, p. 10.

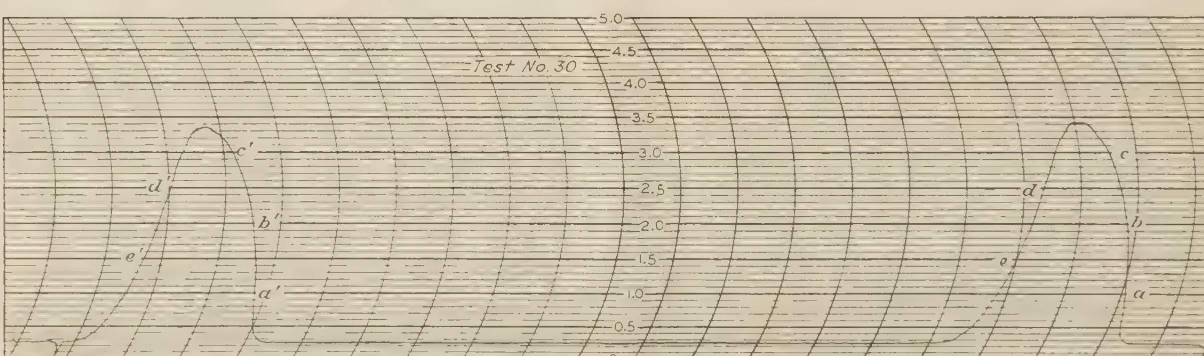

Fig. 7 Recording Chart for Determining Falling Time of the Weights

results of other investigators up to 320 F, which is the upper limit of previous investigations.

The constants for fall-bodies Nos. 35 and 33 which were used in the superheated-steam tests were determined by using dry air at atmospheric pressure and temperatures up to 580 F the viscosity data for the above range having been obtained from the International Critical Tables.[8] Superheated steam up to 500 F at atmospheric pressure was also used for calibrating the constants. The range of pressures and temperatures investigated by the various fall-bodies overlapped sufficiently to obtain a check on the accuracy of the constants.

Test Procedure

In manipulating the apparatus, the fluid in the viscometer was heated to the desired temperature by controlling the electrical input to the heater and was maintained at the desired pressure by manipulating the vent valve or the hand-

[8] International Critical Tables. Published for the National Research Council, McGraw-Hill Book Company, Inc., New York, 1929, vol. 5, p. 2.

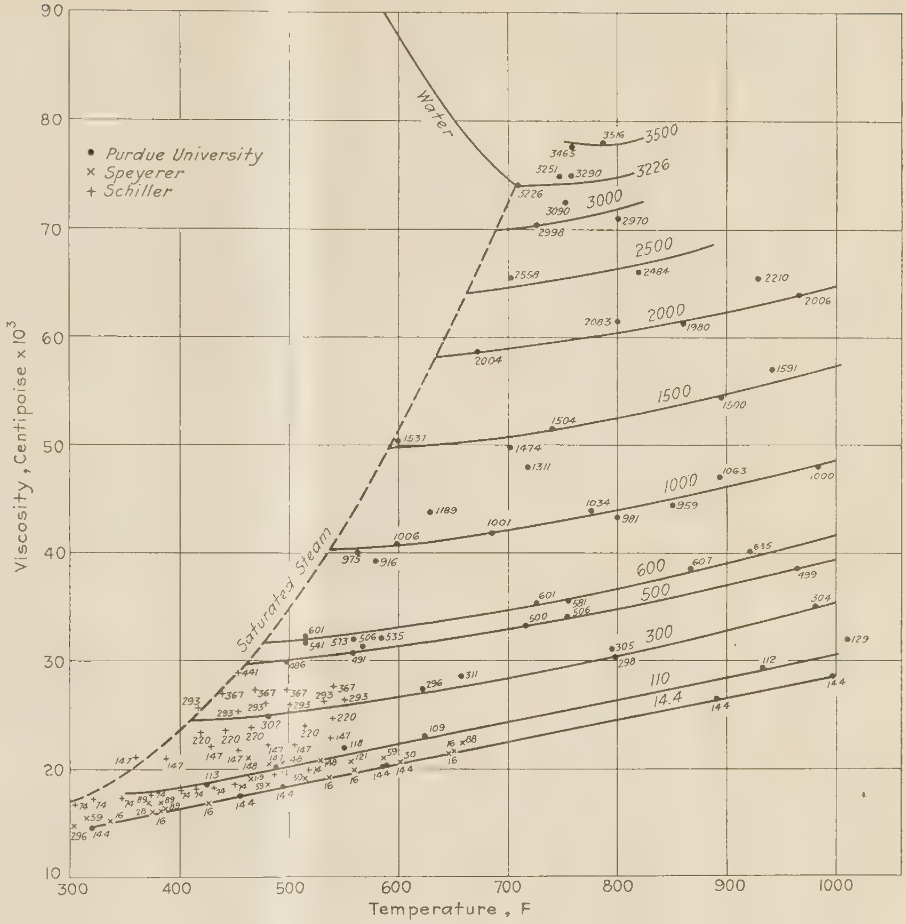

Fig. 9 Viscosity of Water and Superheated Steam
(Small numerals refer to pressure.)

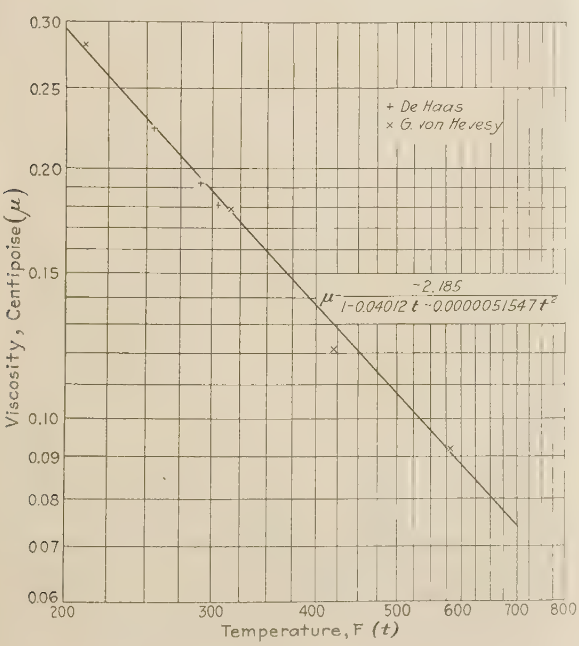

Fig. 8 Viscosity of Saturated Water

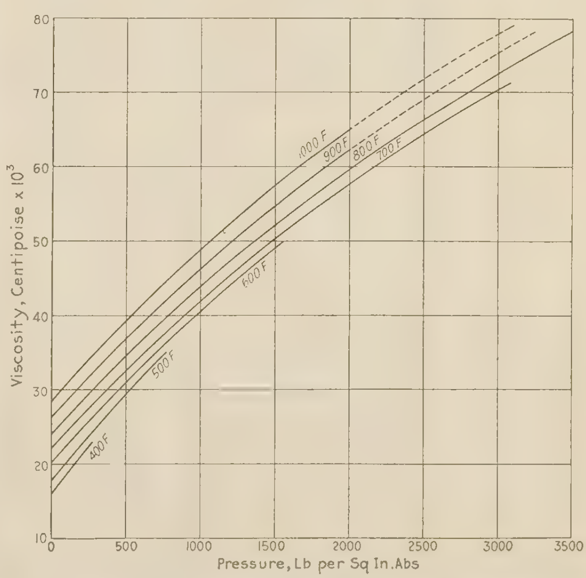

Fig. 10 Effect of Pressure on the Viscosity of Superheated Steam

TABLE 2 VISCOSITY OF SATURATED WATER

Pressure, lb per sq in. abs	Temp, deg F	Viscosity, centipoise
105	331.4	0.170
460	458.5	0.128
850	525.0	0.101
1000	544.6	0.097
1250	572.3	0.092
1505	594.0	0.088
2000	635.6	0.082
2500	668.0	0.078
3000	695.2	0.075
3226	706.1	0.074

TABLE 3 EFFECTS OF PRESSURE ON THE VISCOSITY OF SUB-COOLED WATER

Pressure, lb per sq in. abs	Centipoise at 200 C 392 F	250 C 482 F	300 C 572 F	350 C 662 F	374 C 705 F
500	0.140	...	...	...	..
1000	0.139	0.110	...	...	.
1500	0.138	0.110	0.095	...	..
2000	0.141	0.111	0.096	...	..
2500	0.140	0.110	0.096	0.077	.
3000	0.139	0.109	0.095	0.076	
3226	0.141	0.109	0.095	0.077	0.074
3500	0.140	0.110	0.095	...	...

TABLE 4 TYPICAL DATA ON VISCOSITY OF WATER

Test no.	Pressure, lb per sq in. abs	Temp, deg F	Viscosity, centipoise
515	2495	574	0.096
516	2500	572	0.096
517	2495	572	0.095
518	2500	572	0.096
519	2500	572	0.096
520	2505	574	0.096
521	2500	572	0.096
522	2500	572	0.096
523	2500	572	0.095
Average	2500	572	0.096

TABLE 5 VISCOSITY OF SUPERHEATED STEAM

Pressure, lb per sq in. abs	Temp, deg F	Viscosity, centipoise	Pressure, lb per sq in. abs	Temp, deg F	Viscosity, centipoise
14.4	585	0.0202	975	562	0.0400
14.4	890	0.0265	981	799	0.0432
14.4	997	0.0286	1000	983	0.0480
109	622	0.0230	1001	684	0.0419
111	487	0.0201	1006	598	0.0408
112	932	0.0294	1034	776	0.0438
113	424	0.0186	1063	894	0.0470
118	550	0.0220	1189	628	0.0438
129	1009	0.0320	1311	718	0.0479
296	621	0.0274	1474	702	0.0498
298	799	0.0303	1500	895	0.0545
302	480	0.0249	1504	740	0.0515
304	980	0.0351	1537	598	0.0504
305	794	0.0310	1591	941	0.0571
311	656	0.0286	1980	860	0.0614
486	497	0.0299	2004	672	0.0586
491	558	0.0307	2006	965	0.0641
499	964	0.0386	2083	800	0.0614
506	563	0.0315	2210	928	0.0654
506	567	0.0314	2484	818	0.0661
534	583	0.0320	2558	702	0.0656
541	514	0.0317	....		....
573	558	0.0320	2970	801	0.0710
581	765	0.0355	2998	726	0.0704
601	726	0.0352	3091	752	0.0725
601	514	0.0322	3226	706	0.0740
607	866	0.0384	3251	756	0.0749
635	920	0.0400	3290	746	0.0749
916	579	0.0391	3465	759	0.0775
959	851	0.0445	3516	788	0.0780

operated pump. The switch in the heater circuit was opened and after conditions had reached equilibrium and the readings of the three thermocouples checked within 1 deg, the viscometer was rotated on its trunnions into the vertical position where it was held by a stop. The driving motor of the recording milliammeter was started and the time of fall was recorded automatically on the strip chart. The viscometer was then rotated through 180 deg to permit the fall-body to return to the starting position after which the process was repeated.

The Viscosity of Water

Using fall-body No. 28, the viscosity of saturated water was measured from room temperature to the critical temperature of 706 F. The results are tabulated in Table 2 and plotted in Fig. 8. The effect of pressure on the viscosity of compressed (subcooled) water was determined by increasing the pressure on the water and repeating the test procedure. The results are shown in Table 3. Each tabulated set of values in Tables 2 and 3 is the average of from 5 to 15 individual determinations. A typical set of test results for one pressure and temperature is shown in Table 4. The original data are too voluminous for presentation in this paper, but may be consulted in the office of the Engineering Experiment Station of Purdue University.

Check tests were made over the entire range of pressures and temperatures with fall-bodies Nos. 30 and 31. No difference in results was obtainable with the three fall-bodies.

Keenan's (9) values for the density of compressed water were used in calculating the results.

The viscosity of saturated water between 200 F and the critical temperature may be represented by the empirical equation

$$\mu = -2.185/(1 - 0.04012t - 0.0000051547t^2)$$

where μ is centipoises, and t is degrees F.

The results of this investigation show that at pressures up to 3500 lb per sq in. abs, the effect of pressure on the viscosity of water is negligible, but that viscosity depends upon the temperature.

The Viscosity of Superheated Steam

The viscosity of superheated steam was determined by filling the viscometer with distilled water, then boiling and discharging the excess liquid and vapor through the vent to the atmosphere until the desired pressure and temperature were attained. The instrument was then rotated into the vertical position and clamped, and the time of fall was recorded automatically.

Fall-body No. 35 was used to determine the viscosity of superheated steam for all of the test points while fall-body No. 33 was used to check a number of the previous tests.

The results of these tests are tabulated in Table 5. Each point is the average of from 5 to 40 separate determinations.

Figs. 9 and 10 show graphically the results of the investigation over the entire range of pressures and temperatures from atmospheric pressure to the upper limit of the Keenan steam tables. The authors believe that these results are sufficiently accurate so that they may be used with confidence in engineering calculations.

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Stresses in Three-Dimensional Pipe Bends

By WILLIAM HOVGAARD,[1] NEW YORK, N. Y.

The paper presents first an algebraic method of calculating the couples, forces, and stresses in a three-dimensional pipe line fixed at the ends, consisting of straight parts at right angles to each other connected by quarter bends. The solution is complete in that it takes account of bending, torsion, compression, and shearing. It comprises a special discussion of the rotations and deflections of quarter bends when subjected to forces and couples acting normal to their plane, with particular application to the three-dimensional pipe line. The author also gives an analysis of the stresses at the terminal sections and in the bends, together with a discussion of the strength criteria and the permissible working stresses. A numerical example is given to illustrate the application of the formulas.

1—INTRODUCTION

In a two-dimensional or plane pipe bend subjected to changes of temperature, the reactions at the anchorages are in the plane of the bend. The calculation of the couples and forces acting on any section of the pipe is relatively simple, and has been discussed by the author in four papers[2,3,4,5] published in the Journal of Mathematics and Physics of the Massachusetts Institute of Technology, and in two other papers.[6,7]

When a pipe is so shaped as to lie in three planes, even if at right angles to each other, the problem is more complicated, not only because three coordinates must be used instead of two, but also because torsional couples are produced and the curved parts deflect out of their plane. When the pipe is fixed at the ends, as here assumed, there are at each terminal three component reactions instead of two and three component couples instead of one. Torsional as well as bending stresses are produced and have to be combined with direct and shearing stresses.

When three-dimensional bends are not of simple geometrical form, it is necessary, as in the case of plane bends, to use graphical methods of solution, but when the pipe consists of straight parts and circular quadrants, and provided there are not too many bends and tangents, it is possible and, it is believed, preferable to use the algebraic method, which must always underlie all graphical solutions. It is the object of this paper to present a fairly rigorous solution for the simple case where the pipe has three straight parts and two quarter bends, all situated in three planes at right angles to each other. The method has been successfully applied in practice not only to this case, but also to a pipe having four straight parts and three quarter bends.

It is assumed that the curved parts of the pipe are bent to a radius of curvature not less than four or five times the diameter of the pipe, in which case it is unnecessary to take account of the fact that the neutral axis of a curved bar does not go exactly through the centroid of the section.

On the other hand, we include the effect of the flattening of the circular section which takes place when a curved pipe is bent in its own plane and which may cause a very marked increase in the angular deflection of the bend.[3] This flattening practically does not occur when a curved pipe is bent out of its plane nor is it produced by torsion, so that when calculating the deflections of a pipe bend under such actions, we can regard the pipe as a curved solid bar with the same axis as the pipe and with the same transverse and polar moments of inertia.

The Plane Bend

In developing the following formulas, the general method and notation adopted in the author's paper on deformation of plane pipes[3] are used, but it has been necessary or expedient in dealing with the curved parts of the pipe to extend the method considerably.

Referring first to Fig. 1, let

M	=	a couple acting as indicated by a suffix
R	=	radius of curvature of the axis of a pipe
t	=	$2h$ = wall thickness of the pipe
r_i	=	internal radius of the pipe
r_o	=	external radius of the pipe
r	=	$^1/_2(r_i + r_o)$ = radius of the middle surface of the pipe wall
A	=	sectional area of the transverse section of pipe wall
I	=	moment of inertia of the area of the transverse section about the neutral axis
I_o	=	polar moment of inertia of the transverse section = 2I
E	=	modulus of elasticity
G	=	modulus of rigidity. Taking Poisson's ratio as 0.30, we have: $G = E/2.6$ and $GI_o = EI/1.3$
S	=	any point on the axis of the pipe
ψ	=	angle between the radius to S and some fixed radius of reference

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

$d\psi$ = angle of a small element of the pipe axis comprised between two consecutive transverse sections
$\Delta d\psi$ = change in $d\psi$ caused by flexural strain
s = length or girth of axis from the point of reference to S
ds = $Rd\psi$
K = a coefficient applied to the angular deflection $\Delta d\psi/d\psi$ to allow for the flattening of the transverse section of a curved pipe

$$K = (48h^2R^2 + 10r^4)/(48h^2R^2 + r^4) \ldots\ldots\ldots\ldots [1]$$

Writing

$$2hR/r^2 = \lambda \ldots\ldots\ldots\ldots [2]$$

and substituting λ in Equation [1]

$$K = (12\lambda^2 + 10)/(12\lambda^2 + 1) \ldots\ldots\ldots\ldots [1a]$$

Since the flexibility of a pipe bend is directly proportional to K, this coefficient shall be called the "flexibility factor."

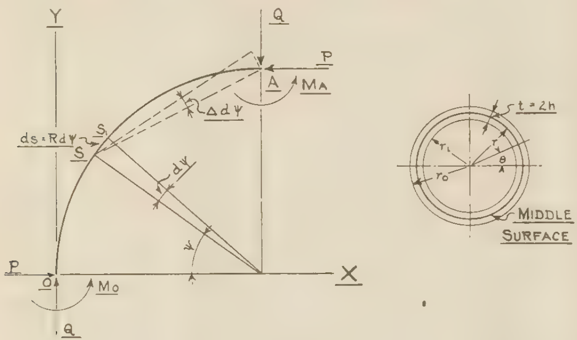

Fig. 1

For brevity, a direction parallel with the X-axis will be called the X-direction, and similarly, the rotation about an axis in the X-direction will be called an X-rotation; a couple with an axis in the X-direction will be called an X-couple; and a linear displacement in the X-direction will be called an X-displacement. Analogous terms are used in connection with the Y and Z axes, an XY-rotation being the rotation in which OX turns toward OY, the same as Z-rotation. The other rotations are similarly referred to. As seen from Fig. 6, a right-handed system of coordinates has been adopted in which the XY-, YZ-, and ZX-rotations are positive.

Axes of couples are indicated in the diagrams by arrows drawn so that when looking along the shaft toward the head, the rotation is clockwise. Thus an arrow drawn in the positive X-direction represents a positive couple and vice versa.

Before dealing with three-dimensional pipe bends, it is desirable first to analyze the case of plane circular quarter bends fixed at one end and subjected to terminal forces and couples acting normal to the plane of the bend at the other end. This problem occurs in three-dimensional bends, and in particular we wish to determine the rotations and the linear displacement normal to the plane of the bend at the free end. The analysis is here carried out in a simple and direct manner, disregarding the displacement of the free end in the plane of the bend, which is of the second order relative to that normal to the plane.

2—QUARTER BENDS UNDER THE ACTION OF FORCES OR COUPLES NORMAL TO THEIR PLANE

Case I—Quarter Bend Fixed at one End and Loaded With a Force Normal to Its Plane at the Free End

Referring to Fig. 2, the bend is represented by its axis, which forms a circular quadrant OA. The bend, which is spoken of as being in a horizontal plane, is fixed at O and loaded by an upward force F at its free end A.

Consider an element ds of the pipe at S. The radius CS forms an angle ψ with the X-axis. The force produces a bending

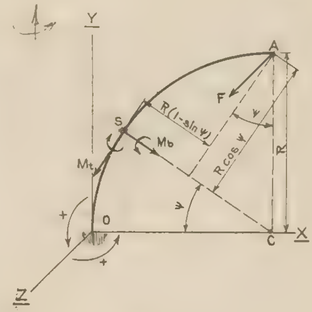

Fig. 2
(Case I.)

moment M_b with axis along the radius at S, and a twisting moment M_t with axis along the tangent at S. These moments can be expressed as

$$\left.\begin{aligned} M_b &= FR\cos\psi \\ M_t &= FR\,(1 - \sin\psi) \end{aligned}\right\} \ldots\ldots\ldots\ldots [3]$$

The moment M_b produces rotation of the element in a vertical tangential plane so that

$$\Delta d\varphi_b = \frac{M_b}{EI}\,ds = \frac{M_bR}{EI}\,d\psi$$

The moment M_t produces a twist of the element

$$\Delta d\varphi_t = \frac{M_t}{GI_o}\,ds = 1.3\,\frac{M_tR}{EI}\,d\psi$$

By resolving these rotations along the X-direction, we obtain for the X-rotation of the element

$$\Delta d\varphi_{xs} = \left[\frac{M_b}{EI}\cos\psi - \frac{1.3\,M_t}{EI}\sin\psi\right]Rd\psi$$

Hence the total rotation at A in the YZ-plane becomes

$$\Delta\varphi_{xA} = \frac{FR^2}{EI}\left[\int^{\frac{\pi}{2}}\cos^2\psi\,d\psi - 1.3\int_0^{\frac{\pi}{2}}(1-\sin\psi)\sin\psi\,d\psi\right] = 0.506\,\frac{FR^2}{EI} \ldots\ldots [4]$$

The Y-rotation of the element is

$$\Delta d\varphi_{ys} = -\left[\frac{M_b}{EI}\sin\psi + \frac{1.3M_t}{EI}\cos\psi\right]Rd\psi$$

and the total rotation of the pipe at A in the XZ-plane becomes

$$\Delta\varphi_{yA} = -\frac{FR^2}{EI}\left[\int_0^{\frac{\pi}{2}}\sin\psi\cos\psi\,d\psi + 1.3\int_0^{\frac{\pi}{2}}\cos\psi\,(1-\sin\psi)\,d\psi\right] = -1.150\,\frac{FR^2}{EI} \ldots [5]$$

It will be noticed that we have here used the couples M_b and M_t and not M_x and M_y, although we wish to find the X-rotation

and Y-rotation. The reason for this is that it is simpler first to find the pure-bending rotation and the pure twist because the corresponding moments of inertia of the pipe section are easily determined. After that we obtain the desired rotations by resolving in the X-direction and the Y-direction. In other words, we resolve rotations instead of couples.

The linear vertical deflection Δz at A is determined as the algebraic sum of that due to M_b and that due to M_t. This deflection can be expressed as

$$\Delta z_A = \int_0^{\frac{\pi}{2}} \frac{M_b}{EI} R \cos\psi \, Rd\psi + 1.3 \int_0^{\frac{\pi}{2}} \frac{M_t}{EI} R\,(1 - \sin\psi)\, Rd\psi$$

$$= \frac{FR^3}{EI}\left[\int_0^{\frac{\pi}{2}} \cos^2\psi \, d\psi + 1.3 \int_0^{\frac{\pi}{2}} (1 - \sin\psi)^2 \, d\psi\right]$$

$$= +1.248 \frac{FR^3}{EI} \ldots\ldots [6]$$

where the positive sign indicates an upward displacement.

Case II—Quarter Bend Fixed at one End and Subjected to a Twisting Moment M_x at the Free End—Fig. 3

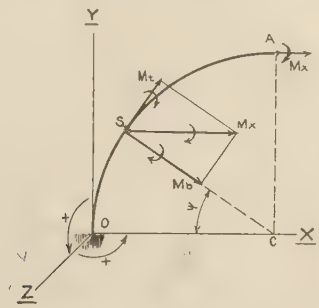

Fig. 3
(Case II.)

From Fig. 3 it is seen that

$$\left.\begin{aligned} M_b &= M_x \cos\psi \\ M_t &= M_x \sin\psi \end{aligned}\right\} \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

The X-rotation of the element is

$$\Delta d\varphi_{xs} = \left[\frac{M_b}{EI}\cos\psi + 1.3 \frac{M_t}{EI}\sin\psi\right] Rd\psi$$

and

$$\Delta\varphi_{xA} = \frac{M_xR}{EI}\left[\int_0^{\frac{\pi}{2}} \cos^2\psi \, d\psi + 1.3 \int_0^{\frac{\pi}{2}} \sin^2\psi \, d\psi\right]$$

$$= +1.806 \frac{M_xR}{EI} \ldots\ldots [8]$$

The Y-rotation of the element is

$$\Delta d\varphi_{ys} = -\left[\frac{M_b}{EI}\sin\psi - 1.3 \frac{M_t}{EI}\cos\psi\right] Rd\psi$$

and

$$\Delta\varphi_{yA} = -\frac{M_xR}{EI}\left[\int_0^{\frac{\pi}{2}} \sin\psi\cos\psi \, d\psi - 1.3 \int_0^{\frac{\pi}{2}} \sin\psi\cos\psi \, d\psi\right]$$

$$= +0.150 \frac{M_xR}{EI} \ldots\ldots [9]$$

It is seen that the major rotation is in the same direction as the acting couple. The Y-rotation is only about 8 per cent of the X-rotation.

The vertical deflection of the free end is

$$\Delta z_A = \int_0^{\frac{\pi}{2}} \frac{M_b}{EI} R \cos\psi \, Rd\psi - 1.3 \int_0^{\frac{\pi}{2}} \frac{M_t}{EI} R(1 - \sin\psi)\, Rd\psi$$

$$= \frac{M_xR^2}{EI}\left[\int_0^{\frac{\pi}{2}} \cos^2\psi \, d\psi - 1.3 \int_0^{\frac{\pi}{2}} (1 - \sin\psi)\sin\psi \, d\psi\right]$$

$$= +0.506 \frac{M_xR^2}{EI} \ldots\ldots [10]$$

where the positive sign again indicates an upward displacement.

Case III—Quarter Bend Fixed at One End and Subjected to a Couple M_y at the Free End, Having Its Axis Normal to the Tangent at A—Fig. 4

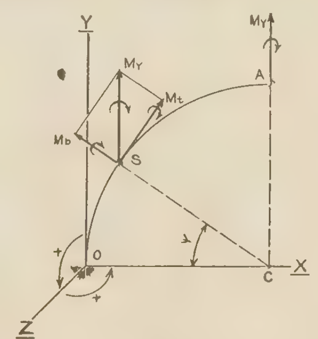

Fig. 4
(Case III.)

It is seen from Fig. 4 that

$$\left.\begin{aligned} M_b &= M_y \sin\psi \\ M_t &= M_y \cos\psi \end{aligned}\right\} \ldots\ldots\ldots\ldots\ldots [11]$$

The rotation of the element in an X-direction at S is

$$\Delta d\varphi_{xs} = \left[-\frac{M_b}{EI}\cos\psi + 1.3 \frac{M_t}{EI}\sin\psi\right] Rd\psi$$

and

$$\Delta\varphi_{xA} = -\frac{M_yR}{EI}\left[\int_0^{\frac{\pi}{2}} \sin\psi\cos\psi \, d\psi - 1.3 \int_0^{\frac{\pi}{2}} \sin\psi\cos\psi \, d\psi\right]$$

$$= +0.150 \frac{M_yR}{EI} \ldots\ldots [12]$$

Rotation of the element in a Y-direction at S is

$$\Delta d\varphi_{ys} = \left[\frac{M_b}{EI}\sin\psi + 1.3 \frac{M_t}{EI}\cos\psi\right] Rd\psi$$

and

$$\Delta\varphi_{yA} = \frac{M_y R}{EI}\left[\int_0^{\frac{\pi}{2}} \sin^2\psi\, d\psi + 1.3\int_0^{\frac{\pi}{2}} \cos^2\psi\, d\psi\right]$$
$$= +1.806\ \frac{M_y R}{EI} \quad \ldots [13]$$

These rotations are the same as found in case II, and we have again a major and minor rotation; but the X-axis and the Y-axis are interchanged.

The vertical deflection is

$$\Delta z_A = -\int_0^{\frac{\pi}{2}} \frac{M_b}{EI} R\cos\psi\, Rd\psi - 1.3\int_0^{\frac{\pi}{2}} \frac{M_t}{EI} R\,(1 - \sin\psi)\, Rd\psi$$
$$= -\frac{M_y}{EI}\left[\int_0^{\frac{\pi}{2}} \sin\psi\cos\psi\, d\psi + 1.3\int_0^{\frac{\pi}{2}} (1 - \sin\psi)\cos\psi\, d\psi\right]$$
$$= -1.150\ \frac{M_y R^2}{EI} \quad \ldots [14]$$

Case IV—Quarter Bend Fixed at One End and Subjected at Any Point S to a Couple Equal to That Which Would Be Produced by a Force F Acting Normal to the Bend at O—Fig. 5

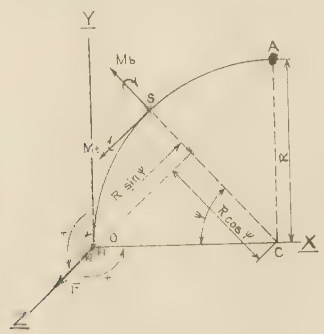

Fig. 5
(Case IV.)

The case is fictitious inasmuch as a force F applied at O would not actually produce a couple at S since O is fixed, but as will be explained later, it is here desired to determine the effect at A of a couple of just that magnitude at any point S. The result is the same as if F were acting downward at A in conjunction with a certain couple.

It is clear that the rotation at the A-end must be of the same amount as in case I, but since the force in case I was applied at the A end, $\Delta\varphi_{xA}$ becomes $\Delta\varphi_{yA}$ in case IV, and vice versa. Thus

$$\Delta\varphi_{xA} = -1.150\ \frac{FR^2}{EI} \quad \ldots [15]$$

$$\Delta\varphi_{yA} = +0.506\ \frac{FR^2}{EI} \quad \ldots [16]$$

Equation [15] gives the twist or X-rotation at A, while Equation [16] gives the Y-rotation at A.

The linear deflection at A normal to the plane will not be the same as in case I. In case IV, $M_b = FR\sin\psi$, and $M_t = FR\,(1 - \cos\psi)$

$$\Delta z_A = -\int_0^{\frac{\pi}{2}} \frac{M_b}{EI} R\cos\psi\, Rd\psi + 1.3\int_0^{\frac{\pi}{2}} \frac{M_t}{EI} R(1 - \sin\psi)\, d\psi$$
$$= -\frac{FR^3}{EI}\left[\int_0^{\frac{\pi}{2}} \sin\psi\cos\psi\, d\psi - 1.3\int_0^{\frac{\pi}{2}} (1 - \sin\psi)(1 - \cos\psi)\, d\psi\right] = -0.408\ \frac{FR^3}{EI} \quad \ldots [17]$$

In cases I to IV, we have supposed the acting force and the axis of the acting couple to be in the positive X- or Y-direction and the axes of all the couples are so drawn as to make the rotation clockwise. The direction in which the arrows representing the couples M_b and M_t are to be drawn can thus be determined readily as can also the sign to be given to the rotations and the displacement at A. If the acting force or couple acts in the negative direction, all the rotations and all the displacements change sign, and we have only in the final results to substitute $-F$, $-M_x$, $-M_y$ for F, M_x, M_y, provided the coordinate axes are placed in the same position relative to the fixed and free ends of the quadrant. The orientation of the quadrant in space is indifferent, but if we change the name of the coordinate axes as in the second quarter bend of Fig. 6, we must realize clearly which end of the quadrant is assumed to be fixed and exactly how the acting force or couple is applied. This is indicated in Fig. 8.

3—ANALYSIS OF THE THREE-DIMENSIONAL PIPE BEND

General

Pipe bends are often attached to parts of the machinery which are anchored at points a certain distance from the terminals of the pipe, these parts of the machinery themselves being subject to heat expansion.

In such cases, linear displacements of the terminals will take place which must be added algebraically to those caused by heat expansion of the pipe itself.

Fig. 6 shows diagrammatically the general lay-out of such a pipe in the ideal case here considered, where all the bends are circular quadrants associated with straight tangents at right angles to each other. The ends O and A are regarded as fixed in direction and subject only to the displacements caused by the machinery parts to which they are attached. Thus, reaction forces and couples exist at those points, and it is the first object of the following analysis to determine these reactions.

Referring to Fig. 6, O is taken as the origin in a right-handed rectangular system of coordinates. The forces and couples $(P,\ Q,\ N)$ and $(M_{Ox},\ M_{Oy},\ M_{Oz})$ at O, and $(-P,\ -Q,\ -N)$ and $(M_{Ax},\ M_{Ay},\ M_{Az})$ at A are the reactions. The forces are considered positive when acting in the positive direction of the coordinate axes and the couples are considered positive when turning in the directions XY, YZ, and ZX. In Fig. 6, all the reaction couples are shown as being positive, but the forces at A, which are known to be equal and opposite to those at O, are shown as being negative.

The radius of the bends is R, and the straight lengths of pipe are L_y, L_x, and L_z, respectively. Also, $L_y + R = H$, $L_x + 2R = L$, and $L_z + R = T$.

The Y-direction shall be referred to as vertical, and the X- and Z-directions as horizontal.

At any section $S\ (x,\ y,\ z)$, the couples acting on the part of the pipe farthest from the origin, but expressed in terms of the reactions at the origin, are

$$\left.\begin{aligned} M_x &= M_{Ox} + Qz - Ny \\ M_y &= M_{Oy} + Nx - Pz \\ M_z &= M_{Oz} + Py - Qx \end{aligned}\right\} \quad \ldots\ldots [18]$$

The couples acting at S on the part of the pipe nearest the origin, expressed in terms of the reactions at A, are of the form

$$M'_x = M_{Ax} + Q(T - z) - N(H - y) = M_{Ax} + QT - NH - Qz + Ny \ldots\ldots [18a]$$

which should be equal to M_x in magnitude, but of opposite sign.

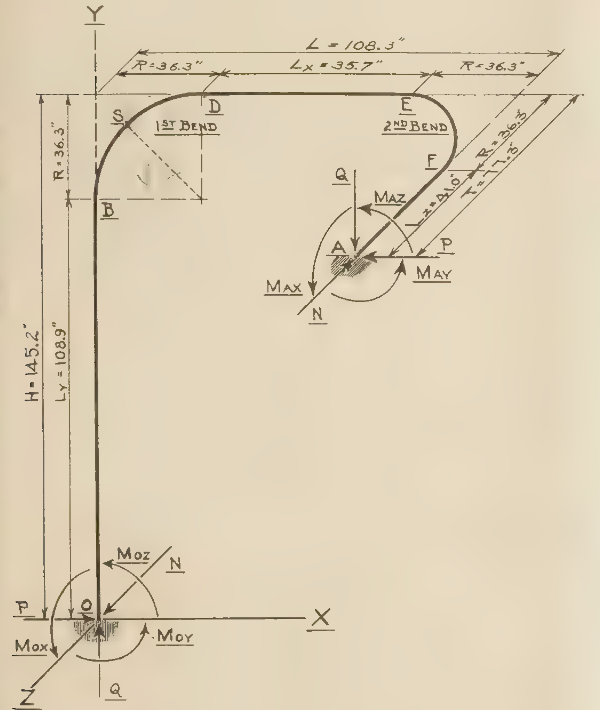

Fig. 6

Now, for equilibrium of the whole pipe taking moments about O, we must have,

$$M_{Ax} + QT - NH + M_{Ox} = 0$$

and substituting in Equation [18a] we get

$$M'_x = -M_{Ox} - Qz + Ny$$

which is equal to $-M_x$ according to the first of Equations [18]. Also M'_y and M'_z, expressed in terms of the reactions at A, are found to be equal to $-M_y$ and M_z, respectively, expressed in terms of the reactions at O.

In calculating the rotations and displacements of the A-end of the pipe, it would seem natural, as in cases I, II, and III, to find the deflections of the element at any point in terms of the reactions at A, that is, in terms of the couples (M'_x, M'_y, M'_z), but since the expressions for those couples, as given by Equation [18a], are more cumbersome than those for (M_x, M_y, M_z) in Equations [18], it is preferred in the following to substitute the latter, simply using $-M_x$, $-M_y$, and $-M_z$ instead of M'_x, M'_y, and M'_z. This consideration led to case IV.

There are six fundamental equations between the reactions and the deflections at each end of the pipe, three for the rotations and three for the linear displacements. Since both ends are assumed to be fixed in direction, the rotations of A relative to O are equal to zero and, therefore, the sign of the bending moments in the equations is indifferent, so that we can use the expressions in Equations [18] as they stand without changing their sign.

In forming the equations for the displacements, it must be borne in mind that the heat expansions at A do not actually take place; they are prevented by the reactions, which may be imagined to produce the displacements Δx_A, Δy_A, Δz_A in negative directions. Hence, while M_x, M_y, M_z must be entered with a negative sign, it is also necessary to enter Δx_A, Δy_A, Δz_A with a negative sign on the other side of the equations, and thus the minus signs cancel one another in this case. (Compare Equations [22] and [23] of this paper with the second and third Equations of [13′] on page 209 of the author's paper on the deformations of plane pipes.[3])

Since the pipe is assumed to be of uniform section and material, the product EI is constant and can be omitted in the equations for the rotations, each of which is equal to zero. In the displacement equations, EI has to be entered, and for the sake of simplicity, is placed as a factor of Δx_A, Δy_A, and Δz_A on the right-hand side of the equations.

It has been shown in case II that a pure or constant couple M_x, applied to a quarter bend, such as for instance the first bend, produces a major rotation (in that case a twist) of the free end in the X-direction, and a minor rotation in the Y-direction. Also, as shown in case III, M_y produces a major rotation in the Y-direction and a minor twist in the X-direction both having the same sign. The minor rotations will be neglected in the equations of rotation, where their inclusion would complicate the problem so as to make the solution impracticable. They can however be included without difficulty in the equations of displacement, and when on this basis the reaction forces and couples have been determined, a more accurate recalculation can be made, if necessary, by substituting these values in the small rotation terms and including them in the rotation equations.

We have seen in cases II and III that a constant couple acting on a quarter bend produces a displacement Δz as well as rotations of the free end. In connection with the three-dimensional pipe we shall refer to that displacement as "local," although the whole pipe beyond the bend partakes in it, but this term is adopted in order to distinguish it from the displacements of A produced by the rotations of the ends of the bends which act with a leverage on the end of the pipe.

Actually the couples, which act on the quarter bends, are not constant throughout the quadrant. Referring to Fig. 7, consider $M_x = M_{Ox} + Qz - Ny$ in the first quarter bend. Here $z = 0$ and $y = L_y + R \sin \psi$, so that M_x is composed of a constant couple $M_{Ox} - NL_y$ and a varying couple $-NR \sin \psi$. Also M_y furnishes a constant couple M_{Oy} and a varying couple $+NR(1 - \cos \psi)$. The effect of the two varying couples is the same as that of a force N acting in the Z-direction at B, since this gives $M_x =$

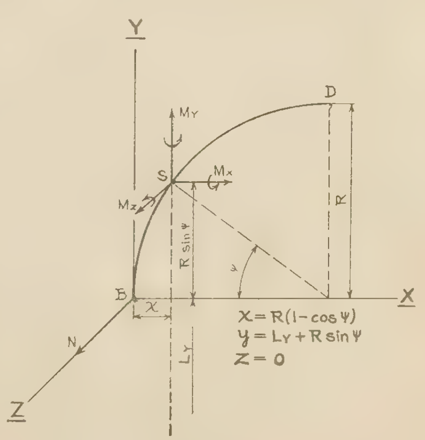

Fig. 7

(First bend, vertical projection.)

$NR \sin \psi$, and $M_y = + NR(1 - \cos \psi)$ and this is precisely the problem considered in case IV where F takes the place of N. We have then the rotations produced by the varying part of the couples M_x and M_y at the point D of the first bend, which are $-1.150\ NR^2$ in the X-direction, and $+0.506\ NR^2$ in the Y-direction.

The displacement Δz produced at D by the force N is, according to case IV, $-0.408NR^3$, which is in the negative Z-direction, and arises from the varying parts of both the M_x and M_y couples. Hence the varying parts, like the constant parts of these couples, produce a local displacement of the quarter bend at D and two rotations.

The M_z couple acts in the plane of the first bend and is dealt with as in a plane problem. Only rotations in the XY-plane are produced, and the displacements at A can be calculated directly by integrating the effect of each element ds as in plane bends.

All that has been said here about the first bend applies in principle to the second bend EF, Fig. 8, where M_y takes the place of

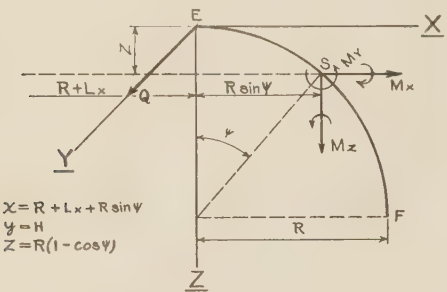

FIG. 8
(Second bend, horizontal projection.)

M_z, and M_x and M_z act in planes normal to the bend. The variable couple is here due to Q acting in the positive direction at E, which is regarded as fixed.

While the straight parts of the pipe can be dealt with by the same general method as used in plane bends, it is expedient to deal with each of the quarter bends separately. The local rotations at the endpoints D and F are determined from cases II, III, and IV and are simply added to the other terms in each of the equations of rotation. In the displacement equations we enter separately the local displacements normal to the plane of each bend and then the displacements caused at A by the rotations at the ends of the quarter bends. This somewhat complicated analysis will be explained in some detail.

In forming the three equations of rotation, the complete integral from O to A for the major component couple is first written, after which are added the integrals for the rotations about the same axis, produced in the quarter bends by the other component couples. While both M_y and M_z produce X-rotation in the first and second bends, respectively, and, therefore, appear in the first equation, M_z does not produce Y-rotation and hence does not appear in the second equation. Similarly M_y does not produce Z-rotation and is therefore absent from the third equation. Thus the equations of rotation become

$$\int_O^A M_x ds + \int_B^D M_y ds + \int_E^F M_z ds = 0 \ldots\ldots [19]$$

$$\int_O^A K M_y ds + \int_B^D M_x ds = 0 \ldots\ldots\ldots\ldots [20]$$

$$\int_O^A K M_z ds + \int_E^F M_x ds = 0 \ldots\ldots\ldots\ldots [21]$$

The integrations are with respect to ds, which in the straight parts of the pipe are equal to dx, dy, dz. In the bends $ds = R d\psi$. The integrations follow the axis of the pipe.

It will be observed that the flexibility factor K is entered only under the integrals where the couple acts in the plane of one of the bends. In all other parts of the system, K is equal to unity. In the first bend, for instance, K is taken into account only in connection with M_z, in the second bend only in connection with M_y.

In forming the displacement equations consider first the X-direction. It is clear that M_x cannot produce any X-displacements in the straight parts and, as explained previously, in the bends except in the first bend where it produces a minor Y-rotation, which with a leverage T causes A to move in the positive X-direction. The couples M_z and M_y must be considered for the whole pipe. The couple M_z acting at any point S, causes the element at that point to rotate and, with a lever $(H - Y)$, to produce a negative X-displacement at A. The couple M_y rotates the element through an angle which, with a lever $(T - z)$, gives a positive X-displacement of A.

The algebraic sum of these displacements will be equal to the heat expansion Δx_A. The equations for the displacements in the Y- and Z-directions are formed in the same manner. Since M_x does not act in the plane of any of the bends, it does not carry the K-factor. The algebraic sums of the displacements are

$$-\int_O^A K(H - y) M_z ds + \int_O^A K(T - z) M_y ds + T \int_B^D M_x ds = EI\, \Delta x_A \ldots\ldots [22]$$

$$\int_O^A K(L - x) M_z ds - \int_O^A (T - z) M_x ds - T \int_B^D M_y ds - L_z \int_E^F M_x ds = EI\, \Delta y_A \ldots\ldots [23]$$

$$\int_O^A (H - y) M_x ds - \int_O^A K(L - x) M_y ds - (L_x + R) \int_B^D M_x ds = EI\, \Delta z_A \ldots\ldots [24]$$

X-ROTATION

The rotation in the YZ-plane is caused chiefly by the first term in Equation [19], containing M_x, which affects all parts of the pipe. The second and third terms of Equation [19], containing M_y and M_z, can produce no rotation in the X-direction in the straight parts of the pipe, and in the bends only where they act normal to the plane of the bends.

Substituting Equations [18] in Equation [19], we obtain

$$\int_O^A (M_{Ox} + Qz - Ny)\, ds + \int_0^{\pi/2} [M_{Oy} + NR(1 - \cos \psi)] R d\psi + \int_0^{\pi/2} [M_{Oz} + PH - Q(R + L_x + R \sin \psi)] R d\psi \ldots\ldots [19a]$$

since $z = 0$ and $x = R(1 - \cos \psi)$ in the first bend, and $y = H$, and $x = (R + L_x + R \sin \psi)$ in the second bend.

For the straight part of the pipe OB, where $z = 0$, we obtain

$$\int_0^{L_y} (M_{Ox} - Ny) dy$$

For the first bend, Fig. 7, where $y = L_y + R \sin \psi$, and $x = R(1 - \cos \psi)$, we obtain

$$\int_0^{\pi/2} [M_{Ox} - NL_y - NR\sin\psi]\,Rd\psi + \int_0^{\pi/2} [M_{Oy} + NR(1-\cos\psi)]\,Rd\psi$$

From case II we know that the constant couple $(M_{Ox} - NL_y)$ produces an X-rotation at A of $+1.806\,(M_{Ox} - NL_y)R$, and from case III, that the couple M_{Oy} produces an X-rotation of $+0.150\,M_{Oy}R$, but as previously stated, the latter will be neglected, at least preliminarily.

There remain the terms

$$-\int_0^{\pi/2} NR^2\sin\psi\,d\psi + \int_0^{\pi/2} NR^2(1-\cos\psi)\,d\psi$$

which are such as would arise from a force N acting normal to the bend at B. This is easily recognized as case IV, where a force F acting at O in Fig. 5 produces an X-rotation at the point A of $-1.150\,FR^2/EI$. Hence we obtain here for the point D a term $-1.150\,NR^2$,

For the straight length $L_x = DE$, extending from $x = R$ to $x = R + L_x$, we still have $z = 0$, but $y = H$. Thus M_x which in this instance is torsional, becomes $(M_{Ox} - NH)$ and we obtain a term

$$+1.30\int_R^{R+L_x}(M_{Ox} - NH)dx$$

The second bend, Fig. 8, is in a horizontal plane and only the first and third integrals in Equation [19a] need to be considered. Here $y = H$, $x = R + L_x + R\sin\psi$, $z = R(1-\cos\psi)$ and we get the terms

$$\int_0^{\frac{\pi}{2}} [M_{Ox} + QR(1-\cos\psi) - NH]Rd\psi + \int_0^{\frac{\pi}{2}} [M_{Oz} + PH - Q(R+L_x) - QR\sin\psi]\,Rd\psi$$

For the constant part of the M_x-couple $(M_{Ox} - NH)$, we get according to case III a major term $+1.806\,(M_{Ox} - NH)R$, and for the constant part of the M_z-couple according to case II a minor X-rotation, $+0.150\,[M_{Oz} + PH - Q(R + L_x)]R$, which again is neglected in the preliminary calculations.

The variable terms in Q give

$$\int_0^{\frac{\pi}{2}} QR^2(1-\cos\psi)\,d\psi - \int_0^{\frac{\pi}{2}} QR^2\sin\psi\,d\psi$$

which are recognized as expressing the effect of a force Q acting upward at E in the same way as F acts at O in case IV, in which $\Delta\varphi_{yA} = +0.506\,FR^2/EI$. Since OY in case IV corresponds to OX in the present case, we get an X-rotation of $+0.506\,QR^2$.

For the length $L_z = FA$, extending from $z = R$ to $z = R + L_z = T$, where again $y = H$, we obtain from the first term in Equation [19a]

$$\int_R^T (M_{Ox} + QZ - NH)dz$$

We can now write Equation [19a] as

$$\int_0^{L_y}(M_{Ox} - Ny)dy + 1.806\,(M_{Ox} - NL_y)R - 1.150\,NR^2 + 1.30\int_R^{R+L_x}(M_{Ox} - NH)dx + 1.806\,(M_{Ox} - NH)R + 0.506\,QR^2 + \int_R^T(M_{Ox} + Qz - NH)dz = 0 \quad [19b]$$

Carrying out the integrations and arranging the terms, we obtain the final equation for the X-rotation

$$[3.612R + 1.30L_x + L_y + L_z]\,M_{Ox} + \tfrac{1}{2}\,T^2Q - [\tfrac{1}{2}\,L_y^2 + 1.806\,(L_y + H)\,R + 1.150R^2 + (1.30L_x + L_z)H]N = 0 \quad [25]$$

If it is desired to allow for the minor rotations caused by M_y and M_z in an eventual recalculation, the following terms

$$0.150\,M_{Oy}\,R + 0.150\,R\,[M_{Oz} + PH - Q(R + L_x)]$$

should be added to the left-hand side of Equation [25]. Substitute the numerical values of M_{Oy}, M_{Oz}, P, and Q as determined by the calculations before these corrections were made; this means simply an addition of a numerical quantity.

Y-Rotation

From Equations [18] and [20] we have

$$\int_O^A K(M_{Oy} + Nx - Pz)ds + \int_0^{\frac{\pi}{2}} [M_{Ox} - N(L_y + R\sin\psi)]\,Rd\psi = 0 \quad [20a]$$

and applying this to the various parts of the pipe we find that

$$1.3\int_0^{L_y} M_{Oy}\,dy + 1.806\,M_{Oy}\,R + 0.506\,NR^2 + \int_R^{R+L_x}(M_{Oy} + Nx)dx + KR\int_0^{\frac{\pi}{2}}[M_{Oy} + N(R + L_x + R\sin\psi) - PR(1-\cos\psi)]\,d\psi + \int_R^T(M_{Oy} + NL - Pz)dz = 0 \quad [20b]$$

The flexibility factor is entered only for the second bend where M_y acts in the plane of the bend.

After integration and transformation Equation [20b] becomes

$$[1.806\,R + 1.571\,KR + L_x + 1.30L_y + Lz]\,M_{Oy} - [0.571\,KR^2 + \tfrac{1}{2}\,T^2 - \tfrac{1}{2}\,R^2]P + [0.506\,R^2 + 2.571\,KR^2 + RL_x + 1.571\,KRL_x + \tfrac{1}{2}\,L_x^2 + LL_z]N = 0 \quad [26]$$

In a recalculation, the numerical value of a term $+\,0.150R\,(M_{Ox} - NL_y)$ should be added to allow for the action of M_x on the first bend (case II).

Z-Rotation

From Equations [18] and [21] we derive the equation

$$\int_O^A K(M_{Oz} + Py - Ox)\,ds + \int_0^{\frac{\pi}{2}}[M_{Ox} + QR(1-\cos\psi) - NH]\,Rd\psi = 0 \quad [21a]$$

from which we obtain

$$\int_0^{L_y}(M_{Oz} + Py)dy + KR\int_0^{\frac{\pi}{2}}[M_{Oz} + P(L_y + R\sin\psi)]\,d\psi + \int_R^{R+L_x}(M_{Oz} + PH - Qx)dx + 1.806\,[M_{Oz} + PH - Q(R + L_x)]\,R - 1.150\,QR^2 + 1.30\int_R^T(M_{Oz} + PH - QL)dz = 0 \quad [21b]$$

The factor K appears in the integral for the first bend, where M_z acts in the plane of the bend. The term $-1.150\,QR^2$ is caused by Q in M_z and M_x in accordance with case IV, where the X-rotation corresponds to the Z-rotation in the present case.

Carrying out the integrations and transposing, we obtain for the Y-rotation

$$(1.806\,R + 1.571\,KR + L_x + L_y + 1.30\,L_z)\,M_{Oz} + (KR^2 + 1.571\,KRL_y + 1.806\,RH + \tfrac{1}{2}\,L_y^2 + HL_x + 1.30\,HL_z)P - (2.956\,R^2 + 2.806\,RL_x + \tfrac{1}{2}\,L^2_x + 1.30\,LL_z)Q = 0 \quad \ldots . [27]$$

In a recalculation, a term $+\ 0.150R\ (M_{Ox} - NH)$ must be added to the left-hand side in order to allow for the action of M_x on the second bend (case III). This is the only term caused by the second integral in Equation [21a].

X-Displacement

Here again the various terms obtained successively from Equation [22] for all the different parts of the pipe are to be added algebraically in order to obtain the displacement of the A-end. From Equations [18] and [22] we obtain

$$-\int_O^A K(H - y)\,(M_{Oz} + Py - Qx)ds + \int_O^A K(T - z)\,(M_{Oy} + Nx - Pz)\,ds + T\int_0^{\frac{\pi}{2}} [M_{Ox} - N(L_y + R\sin\psi)]\,Rd\psi = EI\,\Delta x_A \quad \ldots \ldots [22a]$$

where the last integral represents the displacement due to a minor Y-rotation of the first bend caused by M_x.

The first and second integrals of Equation [22a] run through the whole length of the pipe, but, as explained previously, the bends are treated as independent parts.

In the vertical stem OB, displacements in the X-direction are produced by M_z. At every point, the rotation of an element acts on A with a lever $(H - y)$, giving the term $-\int_0^{L_y} (H - y)\,(M_{Oz} + Py)dy$. At the same time, M_y produces a torsion in OB which at B acts with a lever T relative to the point A, giving a displacement $1.30\,M_{Oy}\,L_y\,T$.

The straight part DE does not produce any X-displacement, but the part FA is bent by M_y and gives a term $+\int_R^T (T - z)M_y\,dz$.

The first bend is subjected to M_z and the second bend to M_y acting in their respective planes, and give rise to two terms, obtained by multiplying the corresponding rotation terms in Equations [21b] and [20b], respectively, with $R\,(1 - \sin\psi)$ and $(T - z)$ under the integral signs. Both terms carry the coefficient K. See Figs. 7 and 8.

In the first bend, the Y-rotation $+\ 1.806\,M_{Oy}\,R$ given in Equation [20b] acts with a lever T. The minor Y-rotation caused by M_x in the third integral of Equation [22a] acts with the same lever.

From the second and third integrals in Equation [22a] there results for the first bend a term which is $+0.506\,NR^2T$. This expresses the effect of the rotation called $\Delta\varphi_{yA}$ in case IV, given in the third term of Equation [20b]. It acts on A with a leverage T.

Thus Equation [22a] becomes

$$-\int_0^{L_y} (H - y)\,(M_{Oz} + Py)\,dy + 1.30\,M_{Oy}\,L_y\,T + \int_R^T (T - z)\,(M_{Oy} + NL - Pz)dz - KR^2\int_0^{\frac{\pi}{2}} (1 - \sin\psi)\,[M_{Oz} + P(L_y + R\sin\psi) - QR\,(1 - \cos\psi)]\,d\psi + KR\int_0^{\frac{\pi}{2}} (L_z + R\cos\psi)\,[M_{Oy} + N(R + L_x + R\sin\psi) - PR(1 - \cos\psi)]\,d\psi + 1.806\,M_{Oy}\,RT + 0.506\,NR^2\,T + 0.150\,(M_{Ox} - NL_y)\,RT = EI\,\Delta x_A \quad \ldots . [22b]$$

Carrying out the integrations and rearranging the terms, we obtain for the X-displacement

$$0.150\,RTM_{Ox} + (1.30\,L_yT + 1.806\,RT + 1.571\,KL_zR + KR^2 + \tfrac{1}{2}\,L^2_z)\,M_{Oy} - (HL_y + 0.571\,KR^2 - \tfrac{1}{2}\,L_y^2)\,M_{Oz} - (\tfrac{1}{2}\,HL_y^2 + 0.571\,KL_yR^2 + 0.430\,KR^3 + \tfrac{1}{6}\,T^3 + \tfrac{1}{3}\,R^3 + 0.571\,KL_zR^2 - \tfrac{1}{3}\,L_y^3 - \tfrac{1}{2}\,TR^2)P + (0.506\,R^2T + 2.571\,KL_zR^2 + 1.50\,KR^3 + KL_xR^2 + 1.571\,KL_xL_zR + \tfrac{1}{2}\,LL_z^2 - 0.150\,RT)\,N = EI\,\Delta x_A \quad \ldots \ldots . [28]$$

Y-Displacement

From Equations [18] and [23] we can derive

$$\int_O^A K(L - x)\,(M_{Oz} + Py - Qx)ds - \int_O^A K(T - z)\,(M_{Ox} + Qz - Ny)ds - T\int_0^{\frac{\pi}{2}} [M_{Oy} + NR\,(1 - \cos\psi)]\,Rd\psi - L_z\int_0^{\frac{\pi}{2}} [M_{Oz} + PH - Q(R + L_x + R\sin\psi)]Rd\psi = EI\,\Delta y_A \quad \ldots . [23a]$$

The first and second integrals of Equation [23a] run through the whole length of the pipe. The third and fourth integrals provide for M_y in the first bend and M_z in the second bend, respectively, which couples act normal to the plane of the bends in both cases.

The displacements caused by the deflections of the straight parts are obtained by applying the proper leverages to the respective X- and Z-rotations as given in Equations [19b] and [21b].

The first bend is subject to Z-rotation in its own plane, acting with a leverage $[L - R(1 - \cos\psi)]$, and causing a Y-displacement at A.

The permanent couples $(M_{Ox} - NL_y)$ in the first bend and $(M_{Ox} - NH)$ in the second bend, which come from the second integral in Equation [23a], cause major rotations in the bends at D and F, respectively, and, as explained in cases II and III, carry the factor 1.806. They act on A with a leverage T and L_z, respectively, causing a negative Y-displacement.

The permanent couples also produce local displacements in the second bend at F. The couple $(M_{Ox} - NH)$, which corresponds to M_y in case III, causes a local Y-displacement of $-1.150\,(M_{Ox} - NH)R^2$, and the couple $(M_{Oz} + PH)$, which corresponds to M_x in case II, causes a local displacement at F of $+0.506\,(M_{Oz} + PH)R^2$.

In the third integral of Equation [23a], M_y causes a minor X-rotation of the first bend, giving a negative term $-0.150RTM_{Oy}$ In the fourth integral, M_z causes a minor X-rotation of the second bend, giving a negative term $-0.150\ RLz\ [M_{Oz} + PH - Q(R + L_x)]$.

The second and third integrals of Equation [23a] give together an X-rotation of the first bend due to the force N acting at B. According to case IV, this rotation is $-1.150\ NR^2$, as given in Equation [19b], and acting at A with a lever T, it gives an upward Y-displacement equal to $+1.150\ NR^2T$.

Similarly the second and fourth integrals of Equation [23a] give together an X-rotation of the second bend at F due to Q acting at E. According to case IV, and as given in Equation [19b], this rotation is equal to $+0.506\ QR^2$, which gives a downward displacement at A equal to $-0.506\ QR^2L_z$. These two integrals also, according to case IV, due to the action of Q, cause a local Y-displacement at F equal to $-0.408\ QR^3$.

We thus arrive at the following equation for the Y-displacement

$$L\int_0^{L_y}(M_{Ox}+Py)dy - T\int_0^{L_y}(M_{Ox}-Ny)dy - 1.30\ L_zT(M_{Ox} - NH) - \int_R^T (T-z)\ (M_{Ox}+Qz-NH)\ dz + \int_R^{R+L_x}(L-x)\ (M_{Oz} + PH - Qx)dx + KR\int_0^{\pi/2}[L-R\ (1-\cos\psi)]\ [M_{Oz} + P(L_y + R\sin\psi) - QR(1-\cos\psi)]d\psi - 1.806\ (M_{Oz} - NL_y)RT - 1.806\ (M_{Ox} - NH)RL_z - 1.150\ (M_{Ox} - NH)R^2 + 0.506\ (M_{Oz} + PH)R^2 + 1.150\ NR^2T - 0.506\ QR^2L_z - 0.408\ QR^3 - 0.150\ RL_z\ [M_{Oz} + PH - Q(R + L_x)] - 0.150\ RTM_{Oy} = EI\,\Delta y_A \quad [23b]$$

After integration and transformation we obtain

$$-(TL_y + 3.612\ L_zR + 2.956\ R^2 + 1.30\ TL_z + \tfrac{1}{2}\ L_z^2)M_{Ox} - 0.150\ RTM_{Oy} + (LL_y + \tfrac{1}{2}\ LL_z + 1.571\ KRL + 0.506\ R^2 - 0.571\ KR^2 - 0.150\ RL_z)\ M_{Oz} + [\tfrac{1}{2}\ LL_y^2 + \tfrac{1}{2}\ HL_x^2 + HLR_z + 1.571\ KRLL_y + KLR^2 - 0.150\ HRL_z - \tfrac{1}{2}\ (K-1)HR^2]P - (\tfrac{1}{6}\ L_x^3 + 0.850\ R^2L_x + 0.850\ RL_x^2 + 0.571\ KLR^2 + 0.408\ R^3 + \tfrac{1}{6}\ L_z^3 + 0.506\ R^2L_z + \tfrac{1}{2}\ RL_z^2 - 0.355\ KR^3)Q + [\tfrac{1}{2}\ TL_y^2 + 1.806\ TRL_y + 1.150\ R^2\ (T + H) + 1.806\ RHL_z + 1.30\ THL_z + \tfrac{1}{2}\ HL_z^2]N = EI\,\Delta y_A \quad [29]$$

Z-Displacement

From Equations [18] and [24] we can derive

$$\int_O^A (H-y)\ (M_{Ox} + Qz - Ny)ds - \int_O^A K(L-x)\ (M_{Oy} + Nx - Pz)ds - (L_x + R)\int_B^D [M_{Ox} - N(L_y + R\sin\psi)]\ Rd\psi = EI\,\Delta z_A \quad [24a]$$

The straight parts OB and DE give rise to the first three terms in Equation [24b], and do not require any explanation.

The couple M_y, acting in the plane of the second bend with a leverage equal to $R\ (1 - \sin\psi)$, carries the factor K. No other Z-displacements are caused by the second bend as can be seen from Fig. 8.

The first bend is acted upon by a permanent couple $(M_{Ox} - NL_y)$ and a varying couple $-NR\sin\psi$. It is also acted upon by a permanent couple M_{Oy} and a varying couple $NR\ (1-\cos\psi)$. The first permanent couple produces a local displacement at D, which according to case II is equal to $+0.506\ (M_{Ox} - NL_y)R^2$. It also produces a Y-rotation acting with a leverage $(L_x + R) = (L - R)$ as given in the third integral of Equation [24a], giving a term $-0.150\ (M_{Ox} - NL_y)\ (L - R)R$. The second permanent couple M_{Oy} gives according to case III a local displacement at D equal to $-1.150 M_{Oy}R^2$, and a Y-rotation with leverage $(L_x + R) = (L - R)$, giving a term $-1.806\ RM_{Oy}\ (L - R)$.

Finally the two varying couples, containing N, give, according to case IV, a local displacement at D equal to $-0.408NR^3$, and since they cause a Y-rotation equal to $+0.506\ NR^2$, they give moreover a negative displacement at A equal to $-0.506\ NR^2\ (L - R)$.

Thus Equation [24a] becomes

$$\int_0^{L_y}(H-y)\ (M_{Ox} - Ny)dy - 1.3\ M_{Oy}\ L_yL - \int_R^{R+L_x}(L-x)\ (M_{Oy} + Nx)dx - KR^2\int_0^{\pi/2}(1-\sin\psi)\ [M_{Oy} + N(R + L_x + R\sin\psi) - PR(1-\cos\psi)]\ d\psi + 0.506\ (M_{Ox} - NL_y)R^2 - 0.150\ (M_{Ox} - NL_y)\ (L-R)R - 1.150\ M_{Oy}R^2 - 1.806\ M_{Oy}R(L-R) - 0.408\ NR^3 - 0.506\ NR^2\ (L-R) = EI\,\Delta z_A \quad [24b]$$

After integration and transformation Equation [24b] becomes

$$(HL_y + 0.356\ R^2 - \tfrac{1}{2}\ L_y^2 - 0.150\ L_xR)M_{Ox} - (1.30\ LL_y + 2.956\ R^2 + 2.806\ RL_x + \tfrac{1}{2}\ L_x^2 + 0.571\ KR^2)M_{Oy} - (\tfrac{1}{2}\ HL_y^2 + 0.914\ R^3 + \tfrac{1}{2}\ LL_x^2 + 1.506\ R^2L_x + 0.785\ KR^3 + 0.571\ KR^2L_x + 0.656\ L_yR^2 - 1/3\ L_x^3 - 1/3\ L_y^3 - 0.150\ LL_y\ R)N = EI\,\Delta z_A \quad [30]$$

Solution of the Equations

The fundamental equations can be written in the following forms

$$A_1M_{Ox} + A_5Q + A_6N = 0 \quad [31]$$

$$B_2M_{Oy} + B_4P + B_6N = 0 \quad [32]$$

$$C_3M_{Oz} + C_4P + C_5Q = 0 \quad [33]$$

$$D_1M_{Ox} + D_2M_{Oy} + D_3M_{Oz} + D_4P + D_6N = EI\,\Delta x_A \quad [34]$$

$$E_1M_{Ox} + E_2M_{Oy} + E_3M_{Oz} + E_4P + E_5Q + E_6N = EI\,\Delta y_A \quad [35]$$

$$F_1M_{Ox} + F_2M_{Oy} + F_6N = EI\,\Delta z_A \quad [36]$$

where the composition of the coefficients A to F with suffixes can be obtained from Equations [25] to [30], inclusive. For a given lay-out of the pipe line, all the quantities that enter into the coefficients are known and their numerical values can be computed.

The solution for the unknown forces P, Q, and N, and for the unknown couples M_{Ox}, M_{Oy}, and M_{Oz} is best obtained by first eliminating the couples. Determine the value of each of the couples from Equations [31], [32], and [33], and substitute the values of the couples in Equations [34], [35], and [36]. This will result in three equations of the form given in the following Equations [37].

$$\left.\begin{aligned}\alpha_1P + \beta_1Q + \gamma_1N &= EI\,\Delta x_A\\ \alpha_2P + \beta_2Q + \gamma_2N &= EI\,\Delta y_A\\ \alpha_3P + \beta_3Q + \gamma_3N &= EI\,\Delta z_A\end{aligned}\right\} \quad [37]$$

from which the unknown forces, P, Q, and N, can be determined.

If these values of the forces so obtained are substituted in Equations [31], [32], and [33], the couples acting at O can be found.

We are now able to determine the couples acting at any point (x, y, or z) of the pipe by substituting the values of the component couples and forces acting at O in Equations [18]. Thus, the couples at the A-end can be found by putting x, y, and z in Equations [18] equal to L, H, and T, respectively. When reversing their sign we have the reaction couples

$$\left.\begin{aligned} M_{Ax} &= -(M_{Ox} + QT - NH) \\ M_{Ay} &= -(M_{Oy} + NL - PT) \\ M_{Az} &= -(M_{Oz} + PH - QL) \end{aligned}\right\} \qquad [38]$$

It is necessary to examine separately the conditions in the bends in order to determine the maximum stresses in these parts. At any point of a bend, one of the three component couples M_x, M_y, M_z acts in the plane of the bend, and stresses caused thereby require the use of the flexibility factor K for their determination. The two other component couples are resolved into one couple M_b with axis in the radius of the bend, causing bending of the pipe. Another couple M_t, with axis tangent to the pipe at the point, constitutes a twisting couple. By means of these couples the bending and torsional stresses can be determined since the corresponding moments of inertia of the section, I and I_o, are readily found.

In the first bend where M_z is acting in the plane of the bend, Fig. 6, the expression for the couple is

$$M_z = M_{Oz} + P\,(L_y + R\sin\psi) - QR(1 - \cos\psi) \qquad [39]$$

Differentiating with respect to ψ, this couple is found to be a maximum when

$$\tan\psi_{\max} = P/Q \qquad [40]$$

Since P and Q are positive, M_z becomes a maximum somewhere within the bend. The maximum value of this couple can be found by substituting from Equation [40] into Equation [39], whence

$$M_{z\max} = M_{Oz} + PL_y + R\left[\sqrt{(P^2 + Q^2)} - Q\right] \qquad [39a]$$

The maximum M_z, however, depending on the sign of M_{Oz}, may be smaller numerically than the couples M_{Bz} and M_{Dz} acting at the points B and D, respectively. The value of M_{Bz} at B where $\psi = 0$ is

$$M_{Bz} = M_{Oz} + PL_y$$

while at D, where $\psi = \pi/2$

$$M_{Dz} = M_{Oz} + PL_y + R(P - Q)$$

It is necessary in all cases to ascertain the value of $M_{z\max}$. The bending couple normal to the plane of the bend is

$$M_b = M_x\cos\psi - M_y\sin\psi = [M_{Ox} - N(L_y + R\sin\psi)]\cos\psi - [M_{Oy} + NR\,(1 - \cos\psi)]\sin\psi$$

Therefore

$$M_b = (M_{Ox} - NL_y)\cos\psi - (M_{Oy} + NR)\sin\psi \qquad [41]$$

The maximum value of M_b occurs when

$$\tan\psi_{b\max} = -\frac{M_{Oy} + NR}{M_{Ox} - NL_y}$$

The twisting couple is

$$M_t = M_x\sin\psi + M_y\cos\psi = (M_{Ox} - NL_y)\sin\psi_{\max} + (M_{Oy} + NR)\cos\psi_{\max} - NR \qquad [42]$$

the maximum value of which occurs when

$$\tan\psi_{b\max} = \frac{M_{Ox} - NL_y}{M_{Oy} + NR} = -\frac{1}{\tan\psi_b}$$

showing that M_t has its maximum in a plane at right angles to M_b. Hence, one of these maxima falls outside the quadrant.

The forces P, Q, and N, acting at any section, are best resolved in directions normal and tangential to the transverse section. They produce direct compression and shear, where

$$F_N = P\sin\psi + Q\cos\psi$$

$$F_s = \sqrt{(P\cos\psi - Q\sin\psi)^2 + N^2}$$

Again it is necessary to determine the maximum values, which for F_N occurs at the same section as that of M_z, while F_s has its greatest value at B and D.

Similar equations can be established for the second bend, where M_y acts in the plane of the bend, and M_x and M_z act at right angles to it.

4—THE STRESSES

Stresses at the Terminal Sections O and A

The greatest stresses are found in the terminal sections and in the bends. The longitudinal bending stress p_{1b} at the O-end of the pipe is generally the most important. This stress is caused by the resultant couple of M_{Ox} and M_{Oz}, here denoted by M_O, where

$$M_O = \sqrt{(M_{Ox}^2 + M_{Oz}^2)} \qquad [45]$$

Since there is no flattening effect, the flexibility factor K is equal to unity, and the corresponding maximum stress is

$$p_{1b} = M_O r_o / I \qquad [46]$$

where r_o is the radius to the outer surface of the pipe.

The steam pressure P produces a longitudinal stress, which is assumed to be uniformly distributed over the section of the pipe wall. This stress is approximately

$$p_{1s} = Pr_i/2t \qquad [47]$$

where r_i is the internal radius of the pipe. This equation is given on page 336 of the author's paper "Tests on High-Pressure Pipe Bends."[5]

Finally, there is a uniform longitudinal stress due to the compressive reaction Q which can be expressed as

$$p_{1c} = Q/A \qquad [48]$$

but which is generally small and can be neglected in many cases.

Thus, the total longitudinal stress becomes

$$p_1 = p_{1b} + p_{1s} + p_{1c} \qquad [49]$$

which has its maximum value at the outer surface in the plane of M_O.

The steam pressure also produces a transverse or "hoop" stress which is assumed to be uniformly distributed over the section of the pipe wall. This stress can be expressed as

$$p_{2s} = Pr_i/t \qquad [50]$$

Where the pipe is straight, as it is at O, no other transverse

stresses exist, so that, denoting the total transverse stress by p_2, we have $p_{2s} = p_2$ in this case.

The torsional couple M_{O_y} produces a shearing stress which has its maximum value in the outer surface. It acts in the plane of the section and is directed everywhere normal to the radius. This stress

$$q_t = M_{O_y} r_o / 2I \qquad [51]$$

A shearing stress q_f is also produced by the resultant of the forces P and N where

$$q_f = \frac{\sqrt{(P^2 + N^2)}}{A} \qquad [52]$$

This stress acts in the same plane as q_t and is added directly to it at the point where the two stresses act in the same direction. It is generally small and can be neglected in many cases. The sum of the shearing stresses produced by the torsional couple M_{O_y} and the resultant of the forces P and N is

$$q_t + q_f = q$$

The stresses at the A-end of the pipe are determined in the same manner.

In addition to the stresses here enumerated, bending and shearing stresses are produced by the deadweight of the piping between hangers and supports. These stresses are to be included in the p_1-stresses and the q-stresses of Equations [49], [51], and [52]. Their values depend on the arrangement of the points of support and are determined by usual methods. With proper design these stresses are generally small and in order not to lengthen the paper they are omitted in the present analysis.

Stresses in the Bends

The curved parts of the pipe require special treatment. The component couple acting in the plane of a bend produces a flattening of the section, causing a peculiar distribution of the longitudinal stresses and the introduction of relatively high transverse stresses. The component couples normal to the plane of a bend produce twisting as well as bending, as previously described in cases I to IV.

In the first bend, that is, the bend nearest the origin, the couple M_z acts in the plane of the bend and has its maximum value as given by Equation [39a] in the section determined by Equation [40]. The stress in the middle surface of the section so determined is generally a maximum, not at the top and bottom of the section, but at points the radius of which forms an angle θ with the neutral axis of the section as shown in Fig. 1. As explained on page 328 of the author's paper "Tests on High-Pressure Pipe Bends,"[5] the maximum value is found by applying a factor β to the ordinary expression for the longitudinal stress, thus

$$p_{1b\max} = \beta(M_{z\max}\, r/I) \qquad [53]$$

where

$$\left.\begin{aligned} \beta &= 2/3\, K \sqrt{[(6\lambda^2 + 5)/18]} && \text{when } \lambda < 1.472 \\ \beta &= (12\lambda^2 - 2)/(12\lambda^2 + 1) && \text{when } \lambda \gtreqless 1.472 \end{aligned}\right\} \qquad [54]$$

The value of λ is given in Equation [2], while the value of K is given in Equation [1a].

Usually $\lambda < 1.472$ and the maximum stress occurs at an angle θ from the neutral axis, such that

$$\sin\theta = \sqrt{[(6\lambda^2 + 5)/18]} \qquad [55]$$

and hence

$$\beta = 2/3\, K \sin\theta \qquad [54a]$$

When $\lambda \gtreqless 1.472$, the maximum value of p_{1b} occurs at the top and bottom of the section.

The value of β varies little from unity except in thin-walled bends of sharp curvature, but θ may be far from a right angle.

The longitudinal bending stresses due to M_b, given by Equation [41], which act in a plane normal to that of the bend, have their maxima at the points which are in the neutral axis relative to M_z, and since $p_{1b\max}$ due to $M_{z\max}$ rarely falls at points less than 45 deg from the neutral axis, these maxima can be considered separately.

The steam pressure produces a longitudinal stress as given by Equation [47], while the reactions P and Q cause a normal stress on the transverse section

$$p_{1c} = (P \sin\psi + Q \cos\psi)/A \qquad [48a]$$

The transverse stress due to the steam pressure is p_{2s} as given by Equation [50], but in addition there is a transverse bending stress p_{2b} due to the flattening of the section with a maximum at $\theta = 0$ deg, and $\theta = 90$ deg, as explained on page 340 of the author's paper "Tests on High-Pressure Pipe Bends."[5] This latter stress is

$$p_{2b} = \frac{18\, M_z \lambda r}{I(12\,\lambda^2 + 1)} \qquad [56]$$

In the outer surface this is a tensile stress, while at the inner surface at the neutral axis for M_z it is a compressive stress, and is reversed in sign at the top and bottom of the section. Compared with the longitudinal stresses in the bend as given by Equation [53], the transverse stresses are very high, and bear a definite ratio to them. If μ be that ratio, then

$$\mu = \frac{p_{2b}}{p_{1b\max}} = \frac{81\,\lambda}{(6\lambda^2 + 5)^{3/2}\sqrt{2}} \qquad [57]$$

For ordinary values of λ, between 0.5 and 1.2, this ratio varies from 1.80 to 1.36, so that the transverse bending stress is on the average 60 per cent greater than the longitudinal bending stress.

There is also a smaller compressive stress in the neutral axis due to the flattening of the section which reduces the tensile stress at the outer surface. It is expressed by

$$p_{2c} = \frac{2Mr^2}{RI}\,\frac{(6\,\lambda^2 + 1)}{(12\,\lambda^2 + 1)} \qquad [58]$$

This equation is the same as Equation [25] on page 83 of the author's paper "The Elastic Deformation of Pipe Bends."[2]

Combining these stresses, the total transverse stress, which will generally be a maximum at the outer surface in the neutral axis, is

$$p_2 = p_{2b} + p_{2s} + p_{2c} \qquad [59]$$

The twisting moment M_x, given by Equation [42], causes a stress, which at the outer surface is given by Equation [51]. The shearing force F_z causes a small shearing stress which generally can be neglected.

The stresses in the second bend are analyzed in the same manner.

Strength Criteria and Working Stresses

Strength criteria and working stresses are among the most important and the most difficult of the pipe-line problems. The difficulty is enhanced by the recent increase in steam pressures and temperatures and by the use of new alloys of steel, with which experience is still very limited. The higher steam pressures produce hoop stresses which can no longer be neglected. At high temperatures the modulus of elasticity and the stress at the yield point are reduced considerably and this, as well as the peculiar nature of each particular alloy, must be taken into account in determining the working stress.

All the stresses so far discussed act in a plane tangential to the cylindrical surface through the point under consideration, and, in combining the stresses, we consider a small element in that plane. If there is no load on the element normal to its plane, we have the first order of a simple case of plane stresses. Referring to Fig. 9, p_1 acts in the X-direction, p_2 in the Y-direction, and the shearing stress q acts on the sides of the element.

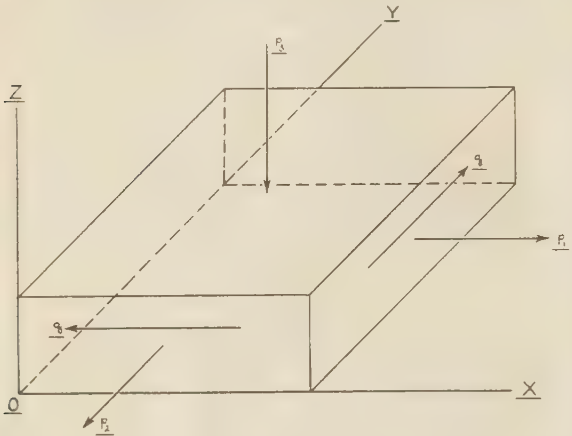

FIG. 9

If we take into account the radial stress p_3, produced by the steam pressure P, we have three-dimensional stress. Actually p_3 is small, being equal to P at the inner surface and decreasing to zero at the outer surface of the pipe wall, but since it has been taken into account by some investigators, it will be included in the following discussion.

For a state of combined stress several strength criteria have been proposed, but we shall only consider the maximum-shear theory and the strain-energy theory,[8] of which the latter appears to be the most rational. Either theory leads to a certain function of the component stresses, so determined that when it is equal in magnitude to the yield point of the material, plastic flow is supposed to occur. This function is referred to as the equivalent stress, which will be noted by p_{eq}. It may or may not represent an actually existing stress, but it is clear that under working conditions the equivalent stress should be smaller than the stress at the yield point of the material.

The Maximum-Shear Theory. According to this theory plastic flow of the material begins to take place when the maximum shearing stress reaches a certain magnitude, regardless of the state of stress. Hence, this stress, must have the same magnitude in the most general case as in the simple case when a straight bar, subject to uniaxial stress, reaches the yield point.

In a state of plane stress, that is, when $p_3 = 0$, or when p_3 is very small, the maximum shearing stress, expressed in terms of the ordinary component stresses, is in the general case

$$q_{max} = \tfrac{1}{2}\sqrt{[4q^2 + (p_1 - p_2)^2]} \qquad [60]$$

In a straight bar subject to a simple tension, $q = 0$, $p_2 = 0$, and therefore $q_{max} = p_1/2$.

When p_1 reaches the yield point p_0,

$$q_{max} = p_0/2 \ldots\ldots [60a]$$

Equating the expressions for q_{max} as given in Equations [60] and [60a], we obtain

$$p_0 = \sqrt{[4q^2 + (p_1 - p_2)^2]} \ldots\ldots [61]$$

The expression on the right-hand side of Equation [61] is the equivalent stress, and is expressed as

$$p_{eq} = \sqrt{[4q^2 + (p_1 - p_2)^2]} \ldots\ldots [62]$$

which, if adopted as the working stress, should then be smaller than p_0.

If we take p_3 into account, we have three-dimensional stress and it is known that in such a case the maximum shearing stress is equal to one-half the difference between the greatest and the smallest of the three principal stress, denoted here by σ_1, σ_2, and σ_3. Let $\sigma_1 > \sigma_2 > \sigma_3$, then

$$q_{max} = (\sigma_1 - \sigma_3)/2 \ldots\ldots [63]$$

Since there are no shearing stresses parallel with the plane of the element, p_3 is a principal stress, and therefore $p_3 = \sigma_3$. The two other principal stresses are found from the formula

$$\left.\begin{matrix}\sigma_1\\ \sigma_2\end{matrix}\right\} = \tfrac{1}{2}(p_1 + p_2) \pm \tfrac{1}{2}\sqrt{[4q^2 + (p_1 - p_2)^2]} \ldots [64]$$

The value of q_{max} from Equation [63] is now equated to its value expressed by Equation [60a] in order to obtain for the equivalent stress

$$p_{eq} = \sigma_1 - \sigma_3 = \tfrac{1}{2}(p_1 + p_2) + \tfrac{1}{2}\sqrt{[4q^2 + (p_1 - p_2)^2]} - p_3 \ldots\ldots [65]$$

This value of the equivalent stress must be smaller than p_0. At the outer surface of the pipe $\sigma_3 = p_3 = 0$, and $p_{eq} = \sigma_1$ which may be greater than p_{eq} as given by Equation [62] for plane stress.

The Strain-Energy Theory. According to this theory, plastic flow occurs when the energy of pure deformation reaches a certain limit, excluding the energy due to change of volume, regardless of the state of stress. Where plane stress occurs, assuming $p_3 = 0$, the energy of deformation per unit volume can be expressed as

$$w = (1/12G)\ [(p_1 - p_2)^2 + p_1^2 + p_2^2] + (1/2G)q^2$$

where p_1, p_2, and q are the total longitudinal, transverse, and shearing stresses, respectively, and G is the modulus of rigidity. Since w must be equal to the energy at the plastic limit under all conditions, it must also be equal to the energy in a bar under simple tension when the yield point is reached. Under these conditions $p_2 = 0$, $q = 0$, and $p_1 = p_0$. Therefore

$$w = p_0^2/6G$$

Equating these expressions for w, the yield point is found to be reached when

$$p_0 = \sqrt{(p_1^2 + p_2^2 - p_1p_2 + 3q^2)}$$

Thus the equivalent stress in this case is

$$p_{eq} = \sqrt{(p_1^2 + p_2^2 - p_1p_2 + 3q^2)} \ldots [66]$$

which, in order to avoid plastic flow, must be smaller than p_0.

If we can disregard q as being insignificant, p_1 and p_2 become principal stresses and Equation [66] takes the form

$$p_{eq} = \sqrt{(p_1^2 + p_2^2 - p_1p_2)} \ldots\ldots [66a]$$

If the transverse stress p_2 can also be disregarded, we have the case of a simple longitudinal stress, where $p_{eq} = p_1$, and therefore $p_1 < p_0$.

In three-dimensional stress, plastic flow will occur according to the strain-energy theory when

$$(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 = 2p_0^2$$

[8] "Zur Theorie plastiche Deformation und der hierdurch im Material hervorgerufenen Nach spannungen," by H. Hencky, *Zeitschrift fur angewandte Mathematik und Mechanik*, vol. 4, no. 4, 1924, pp. 323–334.

Hence

$$p_{eq} = \sqrt{(\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \sigma_1\sigma_2 - \sigma_2\sigma_3 - \sigma_1\sigma_3)} \qquad [67]$$

which should be made smaller than p_0. Here $\sigma_3 = p_3$, while σ_1 and σ_2 can be found from Equation [64].

If σ_3 is neglected

$$p_{eq} = \sqrt{(\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2)}$$

which is identical with Equation [66a], as it should be, since in that equation $p_1 = \sigma_1$, and $p_2 = \sigma_2$.

The Working Stress. The yield point of the material will be used in this paper as the standard of reference, and it is suggested that under working conditions the equivalent stress determined by the strain-energy method as given by Equations [66] or [67] should be not greater than about one-half of the stress at the yield point, that is

$$p_{eq} \lesseqgtr \tfrac{1}{2}\, p_0 \qquad [68]$$

In the author's paper "Tests on High-Pressure Pipe Bends," page 344, it was recommended, on the basis of a number of full-scale tests with pipe bends, to use the longitudinal stress at the middle surface of the pipe wall as strength criterion in the curved parts. It was suggested also that this stress, p_1 in Equation [49] should not be allowed to exceed 16,000 lb per sq in. This means that p_{1b}, the stress due to bending alone, should not be greater than $16{,}000 - (p_{1s} - p_{1c})$, but as previously shown, the transverse bending stresses due to flattening of the bends is much higher than p_1b and might thus become excessive. It was argued, however, that these high stresses were strictly a local phenomenon; they occurred only within very limited regions of the inner and outer surfaces of the pipe wall, and did not seem to affect the general behavior of the pipe. Although that rule led to results which were in accordance with good engineering practice, it appears advisable under more modern conditions of pressure and temperature in all cases to use the equivalent stress as the strength criterion, as recommended previously. If this is done, the transverse stresses will be taken into account, not only in the straight parts of the pipe, but also in the bends. Possibly a somewhat higher working stress may be allowed in the bends than in the straight parts.

These rules are believed to be conservative, but in view of recent rapid developments they must be regarded as suggestions. Before an accurate and fully reliable basis of design can be established, covering all modern conditions, there is required further practical experience over extended periods, and correlated with a theoretical analysis of the stresses. In any case it seems justifiable, in determining the working stress, to take advantage of the higher yield point in alloy steels, while at the same time due account must be taken of the reduction in the stress at the yield point and in the modulus of elasticity at higher temperatures.

5—NUMERICAL EXAMPLE

The numerical analysis for a three-dimensional three-arm pipe, in which the arms are at right angles to each other and connected by right-angle bends, as shown in Fig. 6, is here given to illustrate the application of the formulas presented in this paper. Both ends of the pipe are fixed in position and direction, and when steam is admitted, expansion takes place, not only of the pipe itself but also of certain parts of the machinery to which the ends of the pipe are attached.

The pipe is made of alloy steel, has a modulus of elasticity of 24×10^6 lb per sq in. at 850 F, and a yield point of approximately 39,000 lb per sq in. at 800 F. The temperature rise in the pipe is assumed to be 850 F, at which the expansion of the pipe per inch of length $\delta = 0.006083$ in. Other data are: Working pressure $P =$ 400 lb per sq in.; outer radius of pipe $r_o = 3.644$ in.; inner radius of pipe $r_i = 3.403$ in.; radius of middle surface $r = 3.523$ in.; thickness of pipe wall $2h = t = 0.241$ in.; sectional area of pipe wall $A = 5.33$ sq in.; moment of inertia of A about a diameter $I = 33.14$ in.^{4}; and the radius of pipe bend $R = 36.3$ in. From these data it is found that

$$\lambda = tR/r^2 = 0.705$$

and that

$$K = (12\lambda^2 + 10)/(12\lambda^2 + 1) = 2.29$$

From the dimensions given in Fig. 6, it is observed that $H =$ 145.2 in.; $L_y = 108.9$ in.; $L_x = 35.7$ in.; $L = 108.3$ in.; $L_z =$ 41.0 in.; and $T = 77.3$ in.

The dimensions for expansion are: $H_e = 156.1$ in.; $L_e =$ 96.8 in.; and $T_e = 82.2$ in. Further, $EI\ \Delta x_A = 468.4$; $EI\,\Delta y_A$ $= 755.3$; and $EI\,\Delta z_A = 397.8$.

The coefficients A to F in Equations [31] to [36] are computed from the expressions in Equations [25] to [30].

A sample calculation of the A-coefficients in Equation [25] gives

$3.612\,R$	$= 131.2$	$\frac{1}{2}L_y^2$	$= 5930$
$1.30\,L_x$	$= 46.4$	$1.806\,(L_y + H)R$	$= 16{,}670$
L_y	$= 108.9$	$1.150\,R^2$	$= 1515$
L_z	$= 41.0$	$1.30\,L_xH$	$= 6740$
		L_zH	$= 5955$

$$A_1 = 327.5 \quad A_5 = \tfrac{1}{2}T^2 = 2988 \quad A_6 = -36{,}810$$

The other coefficients are found in the same manner from Equations [26] to [30], inclusive, finally furnishing equations, for this illustration, in which the numerical values of the coefficients replace the algebraic coefficients. The fundamental Equations [31] to [36], inclusive, for this numerical example can now be written with numerical coefficients, which are

$$327.5\,M_{O_x} + 2988\,Q - 36{,}810\,N = 0 \qquad [31]$$

$$427.5\,M_{O_y} - 4223\,P + 20{,}700\,N = 0 \qquad [32]$$

$$407.1\,M_{O_z} + 47{,}330\,P - 13{,}940\,Q = 0 \qquad [33]$$

$$421.0\,M_{O_x} + 26{,}060\,M_{O_y} - 11{,}780\,M_{O_z} - 808{,}000\,P + 956{,}300\,N = 468.4 \times 10^6 \qquad [34]$$

$$-22{,}110\,M_{O_x} - 421.6\,M_{O_y} + 27{,}820\,M_{O_z} + 2{,}798{,}000\,P - 336{,}500\,Q + 2{,}382{,}000\,N = 755.2 \times 10^6 \qquad [35]$$

$$10{,}150\,M_{O_x} - 25{,}390\,M_{O_y} - 791{,}000\,N = 397.8 \times 10^6 \qquad [36]$$

From Equations [31], [32], and [33], the couples can be expressed in terms of the forces, and substituting these values in Equations [34], [35], and [36], we obtain Equations [37] with numerical coefficients, which are

$$\left.\begin{aligned} 817.7\,P - 407.3\,Q - 257.7\,N &= 468.4 \times 10^3 \\ -439.1\,P + 817.8\,Q - 83.1\,N &= 755.3 \times 10^3 \\ -2507\,P - 92.6\,Q + 157.9\,N &= 397.8 \times 10^3 \end{aligned}\right\} \qquad [37]$$

from which the reaction forces $P = 1729$ lb, $Q = 1916$ lb, and N $= 640$ lb, are obtained.

Substituting these values of P, Q, and N in Equations [31], [32], and [33] and solving for the couples, we obtain $M_{O_x} =$ $+54{,}410$ in-lb, $M_{O_y} = -13{,}800$ in-lb, and $M_{O_z} = -135{,}400$ in-lb.

The reaction couples at A as determined from Equation [38] are

$$M_{A_x} = -(M_{O_x} + QT - NH) = -109{,}590 \text{ in-lb}$$

$$M_{Ay} = -(M_{Oy} + NL - PT) = +78{,}280 \text{ in-lb}$$
$$M_{Az} = -(M_{Oz} + PH - QL) = +91{,}900 \text{ in-lb}$$

A corrective calculation was carried out by substituting the values of the forces and couples at O just determined in the expressions for the minor terms omitted in the rotation equations and adding the numerical quantities so found as constants on the left-hand side of these equations. Equations [34], [35], and [36] for the displacements were left unaltered, the minor terms having been included already. The solution of the six equations could now be obtained and were found to give values for the reaction forces and couples which differed from those previously obtained by less than 1 per cent, except that M_{Oz} was —134,670 in-lb, instead of —135,400 in-lb, i.e., about 2 per cent smaller.

Another calculation was made by which the minor terms were omitted in all six fundamental Equations [31] to [36], inclusive, and again the results were practically the same. However, it is recommended to follow the method applied above, by which the minor terms are omitted in the rotation equations but included in the displacement equations.

Determination of Stresses at the O-End

From Equation [45]

$$M_O = \sqrt{(M_{Ox}^2 + M_{Oz}^2)} = \sqrt{(54{,}410^2 + 135{,}400^2)} = 145{,}900 \text{ in-lb}$$

The longitudinal stress at the outer surface due to bending is obtained from Equation [46]

$$p_{1b} = (M_O r_o/I) = (145{,}900 \times 3.644)/33.14 = \pm 16{,}040 \text{ lb per sq in.}$$

The longitudinal stress due to the steam pressure, from Equation [47], is

$$p_{1s} = Pr_i/2t = (400 \times 3.403)/(2 \times 0.241) = +2825 \text{ lb per sq in.}$$

From Equation [48] is obtained the longitudinal stress due to end reaction, which is

$$p_{1c} = Q/A = -1916/5.33 = -359 \text{ lb per sq in.}$$

Hence, the total maximum longitudinal stress at the outer surface, found by applying Equation [49], is

$$p_1 = p_{1b} + p_{1s} + p_{1c} = 16{,}000 + 2825 - 359 = 18{,}500 \text{ lb per sq in.}$$

The hoop stress due to steam pressure, from Equation [51], is

$$p_{2s} = p_2 = Pr_i/t = 400 \times 3.403/0.241 = 5650 \text{ lb per sq in.}$$

The shearing stress due to twisting moment, from Equation [51] is

$$q_t = \frac{M_{Oy} r_o}{2I} = \frac{13{,}880 \times 3.644}{2 \times 33.14} = 763 \text{ lb per sq in.}$$

According to Equation [52], the shearing stress due to the reaction forces P and N are

$$q_f = \frac{\sqrt{(P^2 + N^2)}}{A} = \frac{\sqrt{(1729^2 + 640^2)}}{5.33} = 346 \text{ lb per sq in.}$$

Hence,

$$q = q_t + q_f = 763 + 346 = 1109 \text{ lb per sq in.}$$

The equivalent stress, according to the strain-energy theory is then found by applying Equation [66], where

$$p_{eq} = \sqrt{(p_1^2 + p_2^2 - p_1 p_2 + 3q^2)} = \sqrt{[18{,}500^2 + 5650^2 - (18{,}500 \times 5650) + (3 \times 1109)^2]} = 16{,}530 \text{ lb per sq in.}$$

According to the maximum-shear theory, Equation [62], the equivalent stress is p_{eq} = 13,040 lb per sq in.

Stresses at the A-End

The resultant couple of M_{Az} and M_{Ay} here denoted by M_A, is

$$M_A = \sqrt{M_{Az}^2 + M_{Ay}^2} = \sqrt{(109{,}590^2 + 78280^2)} = 134{,}700 \text{ in-lb}$$

The corresponding maximum stress is

$$p_{1b} = M_A r_o/I = 134{,}700 \times 3.644/33.14 = 14{,}810 \text{ lb per sq in.}$$

The longitudinal stress due to steam pressure is the same as at O, which is 2825 lb per sq in. The uniform longitudinal stress due to the compressive reaction N is

$$p_{1c} = N/A = -640/5.33 = -120 \text{ lb per sq in.}$$

Therefore

$$p_1 = 14{,}810 + 2825 - 120 = 17{,}500 \text{ lb per sq in.}$$

We have again

$$p_2 = p_{2s} = 5650 \text{ lb per sq in.}$$

The shearing stress due to the twisting moment is

$$q_t = \frac{M_{Az} r_o}{2I} = \frac{91{,}900 \times 3.644}{2 \times 33.14} = 5052 \text{ lb per sq in.}$$

While the shearing stress due to the reaction forces at A is

$$q_f = \frac{\sqrt{(P^2 + Q^2)}}{A} = \frac{\sqrt{(1729^2 + 1916^2)}}{5.33} = 484 \text{ lb per sq in.}$$

Therefore

$$q_A = q_t + q_f = 5052 + 484 = 5540 \text{ lb per sq in.}$$

and the equivalent stress, according to the strain-energy theory, Equation [66] becomes

$$p_{eq} = \sqrt{[17{,}500^2 + 5650^2 - (17{,}500 \times 5650) + (3 \times 5540^2)]} = 18{,}200 \text{ lb per sq in.}$$

This still falls below one-half the stress at the yield point, $\frac{1}{2}p_0$ = 19,500 lb per sq in.

According to the maximum-shear theory, Equation [62] p_{eq} = 16,200 lb per sq in., which is again appreciably lower than according to the strain-energy theory.

In most cases the direct stress p_{1c} and the shearing stress q_f can be safely neglected as being inside the limits of accuracy of the computations.

The stresses at O and A are greater than anywhere else in the straight parts of the pipe as can be ascertained by means of Equations [18], so that it remains only to investigate the stresses in the bends.

Stress in the First Bend

In the first bend M_z acts in the plane of the bend and is a maximum when

$$\tan \psi_{max} = P/Q = 1729/1916 = 0.902$$

Therefore ψ = 42 deg 3 min.

The maximum value of M_z as obtained from Equation [39a] is

$$M_{z\max} = M_{Oz} + PL_y + R[\sqrt{(P^2 + Q^2)} - Q] = 135{,}400 + (1729 \times 108.9) + 36.3\,[\sqrt{(1729^2 + 1916^2)} - 1916] = 77{,}000 \text{ in-lb}$$

The bending moment normal to the plane of the bend at any point S as obtained from Equation [41] is

$$M_b = (M_{O_x} - NL_y)\cos\psi - (M_{O_y} + NR)\sin\psi$$

which is a maximum when

$$\tan\psi_{b\max} = -\frac{M_{O_y} + NR}{M_{O_x} - NL_y} = -\frac{-13{,}880 + (640 \times 36.3)}{54{,}410 - (640 \times 108.9)} = 0.613$$

Substituting this value in Equation [41] we find

$$M_{b\max} = (-15{,}280 \times 0.852) - (9360 \times 0.523) = -17{,}900 \text{ in-lb}$$

The twisting moment at S, according to Equation [42], is

$$M_t = (M_{O_x} - NL_y)\sin\psi + (M_{O_y} + NR)\cos\psi - NR$$

which is a maximum when

$$\tan\psi_{t\max} = -1/\tan\psi_{b\max}$$

This places the maximum outside the quadrant and we have to determine the value of M_t where $\psi = 0$, and $\psi = \pi/2$. It is found that for $\psi = 0$, $M_t = M_{O_y} = -13{,}880$ in-lb, and for $\psi = \pi/2$, $M_t = 38{,}520$ in-lb. Hence, M_t has its greatest value at D, and since M_z has its maximum at $\psi = 42$ deg, and M_b at $31^1/_2$ deg, and since they are all small or moderate as compared with the couples at A and O, it is unnecessary to consider any combination of the three couples. It is of interest, however, to investigate the stress caused by $M_{z\max}$, which is found from Equation [53].

The maximum stress occurs at a point in the section, the radius of which forms a certain angle θ with the neutral axis. This angle can be found from Equation [55], in which $\lambda = 0.705$. Therefore

$$\sin\theta = \sqrt{[(2.982 + 5)/18]} = 0.666$$

and since $K = 2.29$, the value of β can be found from Equation [54a], where

$$\beta = 2/3\, K \sin\theta = 2/3 \times 2.29 \times 0.666 = 1.017$$

The maximum stress can now be determined from Equation [53], where

$$p_{1b\max} = (1.017 \times 77{,}000 \times 3.523)/33.14 = 8320 \text{ lb per sq in.}$$

It is unnecessary further to peruse the study of the longitudinal stress in this case, since it is known to be much smaller than the transverse stress. From Equation [56] we have at the outer surface in the neutral axis

$$p_{2b} = \frac{18\, M_z \lambda r}{I\,(12\,\lambda^2 + 1)} = \frac{18 \times 77000 \times 0.705 \times 3.523}{33.14\,(12 \times 0.497 + 1)} = 14{,}930 \text{ lb per sq in.}$$

The hoop stress p_{2s} due to steam pressure was found previously to be 5650 lb per sq in. The transverse compressive stress due to flattening, applying Equation [58], is

$$p_{2c} = \frac{2M_z r^2}{RI}\,\frac{(6\,\lambda^2 + 1)}{(12\,\lambda^2 + 1)} = \frac{2 \times 77000 \times 12.41}{36.3 \times 33.14} \times \frac{3.982}{6.964} = 910 \text{ lb per sq in.}$$

Hence, from Equation [59] the total transverse stress in the neutral axis is

$$p_2 = 14{,}930 + 5650 - 910 = 19{,}670 \text{ lb per sq in.}$$

At the section where $M_{z\max}$ is acting, there is a twisting moment as given by Equation [42]. With $\psi_{\max} = 42$ deg 3 min, we find after applying Equation [42] that $M_t = 26{,}520$ in-lb. Hence, a shearing stress is produced at the outer surface which, according toEquation [51], is found to be

$$q_t = \frac{M_t r_o}{2I} = \frac{(26{,}520 \times 3.644)}{(2 \times 33.14)} = 1460 \text{ lb per sq in.}$$

The equivalent stress in the neutral axis with $p_1 = 0$ and neglecting p_3, from Equation [66] is

$$p_{eq} = \sqrt{(p_2^2 + 3q^2)} = \sqrt{[19{,}670^2 + (3 \times 1460^2)]} = 19{,}900 \text{ lb per sq in.}$$

which is a little more than one-half the stress at the yield point.

The investigation of the stresses in the second bend are made in the same manner.

6—ACKNOWLEDGMENT

The author wishes to express his indebtedness to Henry C. E. Meyer, chief engineer of Gibbs and Cox Inc. for his advice in the preparation of this paper; in particular, the discussion of the permissible working stress. His thanks are due also to Horace L. Bickford of Gibbs and Cox Inc. for his assistance in the preparation of the diagrams as well as in working out the numerical example.

An Airfoil-Type Propeller Blower for Ventilating Electrical Machines

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The authors discuss the development of an airfoil-type propeller blower, the design of which was influenced by space limitations in connection with the motor-driven blower equipment for a 165,000-kw turbine-generator unit installed at the Richmond station of the Philadelphia Electric Company. After the fan was designed, a model was built and tested before the full-size equipment was constructed. The authors present performance data for both the model and the full-size blower.

NEW types of apparatus of improved design and performance are frequently developed as a result of an effort to overcome certain difficulties in a particular commercial application. Space limitations in connection with the motor-driven blower equipment for a large turbine-generator unit brought about the development of the blower described in this paper. When the latest addition to Richmond station of the Philadelphia Electric Company was planned, it was decided to add a 165,000-kw turbine-generator unit. As the turbine-room structure was already in existence with space originally designed for four 50,000-kw units, it will be seen that the design of a 165,000-kw unit to go in the space originally occupied by a 50,000-kw unit involved a careful use of all the available space. The specifications called for all blowers, air coolers, and air ducts to be located below the generator. The height of the space available for this equipment was fixed by the height of the existing turbine-floor level which was 28 ft 6 in. above the condenser-floor level. The width of the space was limited on one side by a circulating pump and on the other by an elevated track. Space had to be provided to permit ready removal of air-cooler tubes. The specifications called for four motor-driven blowers, one of which was to be a spare unit. The generator required a maximum volume of 170,000 cfm. The calculated pressure drop in the generator air coolers and ducts was 12 in. of water. Therefore, the volume rating of each blower was to be 57,000 cfm.

A study of the space available under the generator soon indicated that the usual types of blowers available would occupy too much space and would not fit in with any reasonable system of air ducts. Having had considerable experience with propeller blowers built on the shafts of turbine generators, it was decided to undertake the development of a propeller-type blower of high speed and small dimensions to suit this application.

Another requirement to be met by these blowers was that they were to be suitable for full-voltage starting control for the induction motors driving them. As the generator would operate at all times with two or three blowers running in parallel, the blower characteristics had to be such that good parallel operation would be obtained. As the blowers were to be part of a closed ventilating system in which the air was returned to the generator after being cooled in finned-tube coolers, it was necessary to provide them with suitable intake ducts. Air leaving the generator and entering the blower may reach a temperature of 150 F and some means had to be provided in the design to avoid excessive heating of the blower bearings with this high-temperature air adjacent to them. The speeds available for the blowers corresponded to the standard speeds for 60-cycle induction motors. The speed adopted was 3550 rpm.

Description of Full-Size Blower

The design of the full-size blower is shown in section in Fig. 1. The fan runner consists of eight aluminum-alloy blades dovetailed into a forged-steel hub. The fan runner is overhung on two sleeve bearings mounted in a common rigid cast-iron pedestal as shown in Fig. 2. The shaft and bearings were designed on the premise that no serious damage would occur to these parts in the event of losing one blower blade at full speed. The provision of a rigid shaft and pedestal contributed largely to the smooth running of the unit as a whole as found by test. The thrust varies with the back pressures against which the blower is working. In order to take care of the highest thrust obtained, a two-shoe Kingsbury thrust bearing was incorporated in the same housing as the journal bearing adjacent to the motor coupling.

As the blower bearing losses amounted to something over 1 kw, a small centrifugal fan was mounted on the blower shaft to force a stream of air through passages under the oil chamber in the pedestal, thus cooling the lubricating oil. This cooling air discharges into the space between the pedestal and the blower casing and effectively prevents heat from the casing affecting the pedestal. To provide circulation of the lubricating oil, a small oil pump is mounted on the outboard-motor bracket and connected to the blower pedestal by brass piping.

Air from the generator enters the blower casing from the top through a hand-operated damper which can be closed when a blower is shut down with other blowers in operation. The blower casing is constructed of welded-steel plate sufficiently rigid to support the guide-vane and diffuser structure which is bolted with a flanged joint to the casing. Air enters the blower through a spun-aluminum intake nozzle, passes through the blower and then through the stationary guide vanes in the diffuser. The guide vanes are built up of aluminum castings.

Theoretical Design of the Fan

The fan without guide vanes and diffuser was designed on the basis of the theory outlined in a paper by Dr. O. G. Tietjens.[3] Within certain empirical limits the proportions of the fan itself,

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

[3] "The Propeller-Type Fan," by O. G. Tietjens, Trans. A.S.M.E., vol. 54, 1932, paper APM-54-13, pp. 143–152.

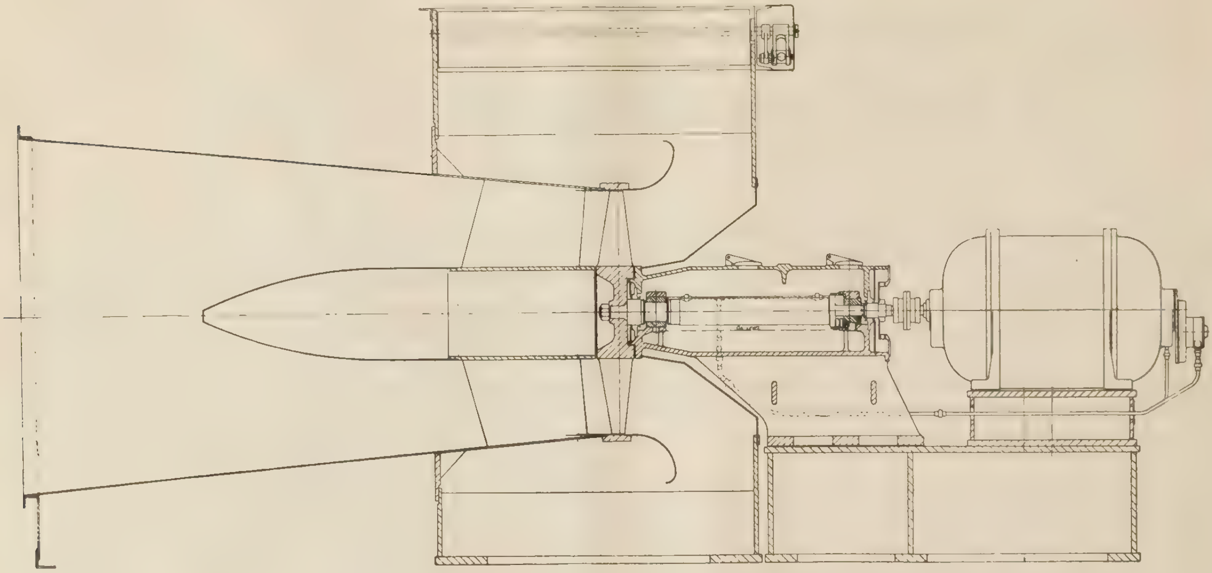

FIG. 1 CROSS-SECTION ASSEMBLY OF THE FULL-SIZE UNIT

without guide vanes or diffuser, can be accurately predetermined for a given pressure and volume. Certain empirical limits applied to the theoretical design had been found by means of considerable experimental investigation performed by H. C. Werner of the Westinghouse Research Laboratories on several different fans of the air-foil type. The guide vanes and diffuser add considerably to the capacity and efficiency of the fan. As a result, the theoretical design of the fan itself was based on 90 per cent of the required volume of 57,000 cfm per fan and 85 per cent of the required static pressure of 12 in. of water, which is approximately 25 per cent lower than the required capacity. It was assumed that the diffuser would recover between 70 and 80 per cent of the dynamic energy created by the fan in a direction parallel with the axis of rotation. In addition, it was assumed that the guide vanes would recover about 50 per cent of the rotational energy at the discharge side of the fan. How closely the tests verified these assumptions is discussed later.

The fan blade has an airfoil cross-section of varying profile from hub to tip. The general shape is that of a cross-section of an airplane wing having a flat side and well-rounded leading edge which is stable for large angles of attack. The blade width and angle with respect to a vertical plane gradually decrease from hub to tip. The fan contains eight blades so spaced at the hub that no overlapping exists. Mutual interference of one blade upon another was found to be very small with this spacing.

In line with the theory outlined in Tietjens' paper,[3] the following is a brief analysis of the factors to be considered in the theoretical design: In designing a propeller fan the total pressure generated by the fan multiplied by the volume rate of flow represents a certain power output. The volume rate of flow is equal to the fan area times the axial velocity through the fan. Hence, the power output is

$$Ap_t w = Tw \qquad [1]$$

where A is the fan area, w is the axial velocity through the fan, p_t is the total pressure (static plus dynamic), and T is the total force acting on the fan area and is called the thrust. In general, the problem reduces to that of designing a fan which will produce the total thrust corresponding to the required total pressure.

From airfoil and propeller theory, the thrust depends on the magnitude and distribution of the aerodynamic circulation around the blade. This may be given by the relation

$$dT = (n\rho\ \Gamma\ r\omega)\ dr \qquad [2]$$

where dT is the incremental thrust at a point corresponding to radius r on the blade, Γ is the circulation, ω is the angular velocity of the fan, n is the number of blades, and ρ is the air

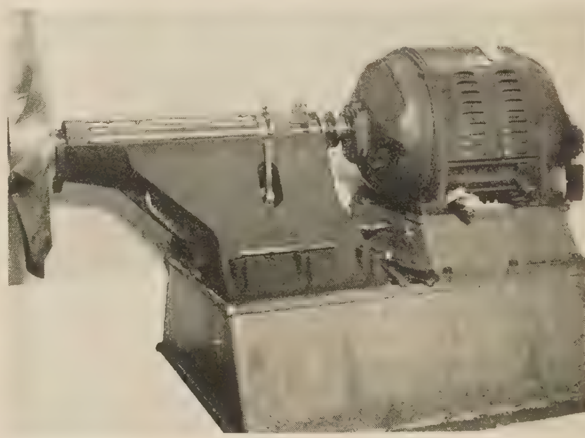

FIG. 2 FAN RUNNER AND BEARING OF THE AIRFOIL BLOWER

density. It has been shown theoretically that the pressure rise through a propeller fan and, hence, the thrust, can only be obtained by the creation of a rotational-velocity component which can be clearly observed at the discharge side of the fan. The circulation and rotation are related to the blade dimensions at radius r by the equation

$$\Gamma = C_L\ bv/2 \qquad [3]$$

in which C_L is the coefficient of lift of the blade at the air-foil cross-section corresponding to the radius r; b is the width of the blade; and v is the velocity of air relative to the blade. The relation between v and the axial velocity through the fan and the peripheral

speed at the radius r and the rotational component at the radius r is shown geometrically in Fig. 3. The angle of attack α is necessary to produce the required lift at radius r, and $\alpha + \beta$ is the angle at which the blade is located on the hub.

The fundamental definition of the term circulation is given by the relation of the quantities defined in Equation [3]. Physically, the circulation is an idealized component of the actual flow around an airfoil which can be represented by the streamlines completely traversing the airfoil in a rotating manner. A detailed explanation of the meaning of circulation is given by Prandtl-Tietjens.[4]

If the circulation were assumed constant over the entire blade area in Equation [2], the thrust would vary directly with the radius only, and the total thrust could be obtained by analytical integration of Equation [2]. However, with constant circulation the portions of the blade at the smaller radii are worked at relatively high outputs and the rotation losses become much higher than if the circulation were decreased with decreasing blade radius. It is possible to have constant circulation over the blade area without large rotation losses if the hub diameter is made large relative to the outside diameter, which has the effect of increasing the outside diameter. For a given required total thrust, a compromise is reached between the size of hub relative to outside diameter.

The circulation distribution is decreased toward the hub in such a way that rotation losses are reduced to a value consistent with high efficiency. The fan considered in this paper would have had 45 per cent more rotation loss if the circulation had been constant over the blade area, which is equivalent to a reduction of 5 per cent in the fan efficiency. The ratio of hub diameter to outside diameter was 0.4. This rotation loss is the component of energy at the fan discharge which lies in a plane perpendicular to the axis of rotation of the fan. The dynamic energy is the axial component of the energy at the fan discharge.

Table 1 gives the important constants of the fan referred to in these tests. The volume and pressure are for the fan itself without guide vanes or diffuser at the calculated design point which is also the point of maximum efficiency.

TABLE 1 PRINCIPAL CONSTANTS OF THE AIRFOIL PROPELLER BLOWER

Air density, lb per cu ft	0.0749
Volume, cfm	52,500
Static pressure, in. of water	10.2
Speed, rpm	3600
Outside diameter, in.	38.31
Inside diameter, in.	14.50
Fan area, sq ft	6.87
Constant axial velocity through fan, cfm	7650
Dynamic pressure head, in. of water	3.55
Total head, in. of water	13.75
Ratio of dynamic to total head	0.258
Number of blades	8

The ratio of dynamic head to total head is an important constant for the propeller-type fan because if this value is made too low, such as 0.10 to 0.15, the application of airfoil theory to the fan design begins to become inaccurate because the nature of the flow around the blades is changing due to centrifugal effects caused by the air rotating much faster at the lower-volume points. Values of the ratio between 0.20 and 0.30 have been found satisfactory from the standpoint of performance and efficiency.

Description of Model-Test Set-Up

The model was built about one-quarter size and geometrically similar to the proposed full-size application in all the important details including the inlet box, inlet guides, and diffuser. The best proportions of such parts as the guide vanes and diffuser as determined from the model tests were incorporated in the final full-size unit with exact geometric similarity.

A schematic diagram of the model-test set-up is shown in Fig. 4. The fan draws air through a long gradually diverging inlet duct at the entrance of which is located a flow nozzle for measurement of air volumes. The inlet duct is fastened to the side of the inlet box of the fan. The fan discharges air directly to the room or into the diffuser and guide-vane arrangement. The fan shaft extends through the inlet box and is coupled to a $^1/_2$-hp electric dynamometer used to measure the power input to the fan and also drive the fan. The duct ahead of the flow nozzle was provided with a well-rounded entrance in the form of a one-piece smooth spun-copper ring. Between this ring and the duct

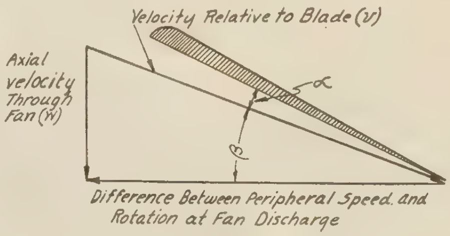

Fig. 3 Air Velocity Relative to the Blade

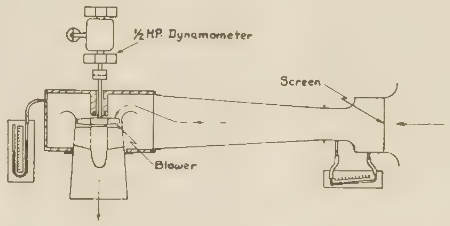

Fig. 4 Diagram of Test Arrangement for the Model Blower

flange, screens were placed in order to vary the pressure-volume delivery of the fan.

Measurements

The volume of air was measured by means of a flown ozzle made of smooth spun copper. The proportions of the nozzle were similar to one of those built by the Bureau of Standards and calibrated for use in calibrating orifices.[5] An average nozzle coefficient of 0.980 was used. The pressure drop across the nozzle was measured by means of eight accurately constructed static-pressure taps on the inlet and discharge sides of the flow nozzle. The pressure drops were measured on an inclined manometer the scale of which was such that errors were within $^1/_2$ per cent for normal fan delivery.

The static pressure developed by the fan was measured by means of taps in the top corners of the inlet box. The average velocity to the inlet box was about 500 fpm which is low enough to introduce negligible error in the measurement of static pressures from $^1/_2$ to 2.5 in. of water found in the model tests.

The power input was measured by means of a dynamometer arrangement consisting of a $^1/_2$-hp direct-current motor used to drive the fan. The stator of the dynamometer was suspended in ball bearings and equipped with a lever arm and scale pan. The torque reaction was measured by balancing it against weights

[4] "Applied Hydro- and Aeromechanics," by O. G. Tietjens, based on lectures of L. Prandtl, McGraw-Hill Book Company, Inc., New York, 1934.

[5] "Discharge Coefficients of Square Edged Orifice for Measuring the Flow of Air," by H. S. Bean, E. Buckingham, and P. S. Murphy, *Bureau of Standards Journal of Research*, March, 1929, pp. 568–652.

placed in the scale pan mounted on the lever arm. Speed was maintained at a given value by using current supplied from a separately excited direct-current generator driven by a single-phase alternating-current motor. The sensitivity of the suspension was such that the average input of $^1/_4$ to $^1/_3$ hp could be checked within 1 per cent. Extreme care was exercised to obtain accurate results. Checks were obtained by means of repeat runs after a considerable time interval. All data were corrected to the standard air density of 0.07488 lb per cu ft. On repeat runs, the pressure multiplied by the volume and divided by the power input which measures the fan efficiency could be checked within 2 per cent.

Model Tests and Test Results

Tests on Fan Itself. In all of the model tests made, the intake box was in place. All of the model data have been expressed in terms of the values for the full-size unit. The results of the tests of the fan itself without guide vanes or diffuser are shown in Fig. 5. The test-performance curve checks closely with the working point for which the fan was designed. The static efficiency of 56 per cent at the working point is the approximate maximum static efficiency. The total efficiency of the fan itself without guide vanes or diffuser is about 75.8 per cent and is based on the static pressure plus the dynamic pressure corresponding to the average axial-velocity component of the air at the fan-discharge area. The difference between 100 per cent and the total efficiency

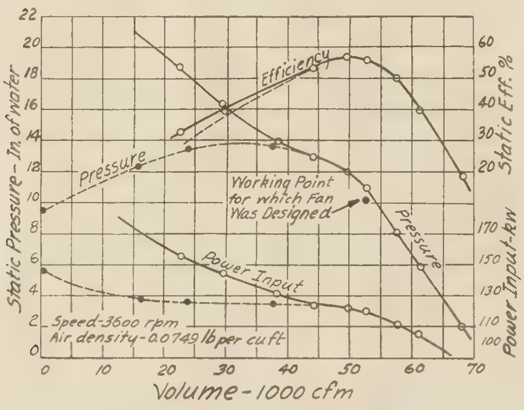

Fig. 5 Performance Curves of the Fan Alone
(Performance with inlet-box baffles removed is shown by dashed lines while performance with baffles in the inlet box is shown by solid lines.)

is a measure of the sum of the fan losses expressed in terms of per cent efficiency. A detail discussion of the magnitude and distribution of these losses is given later.

The solid curve in Fig. 5 represents the performance with the intake box in which were placed two vertical baffle plates diametrically opposite. These plates were fitted around the curved inlet guide which is part of the fan housing. With these plates in place, the pressure characteristic was rising with decreasing volume which is a necessary property for satisfactory operation of two or more fans in parallel. When the plates were omitted from the intake box, the pressure characteristic assumed the "hump" shape which is represented by the dashed line in Fig. 5. This shape is not suitable for parallel operation. The performance with and without the plates was the same at the higher-volume points and in the region of the desired working point. The "hump" shape is caused by the air whirl induced in the inlet box which becomes large enough in magnitude at the lower volumes to reduce the static pressure considerably below the values obtained when no such whirl is present.

The flat portion on the fan curve is characteristic of the propeller-type fan. From a theoretical standpoint, this sudden change in the slope of the pressure characteristic is caused by too large an angle of attack on the blade section. From airfoil theory, as the angle of attack increases, a point is reached at which the lift becomes a maximum and any further increase in angle of attack results in a sudden decrease in the lift, the flow conditions around the blade becoming unstable. In the propeller fan, as the axial velocity through the fan decreases the angle of attack at a

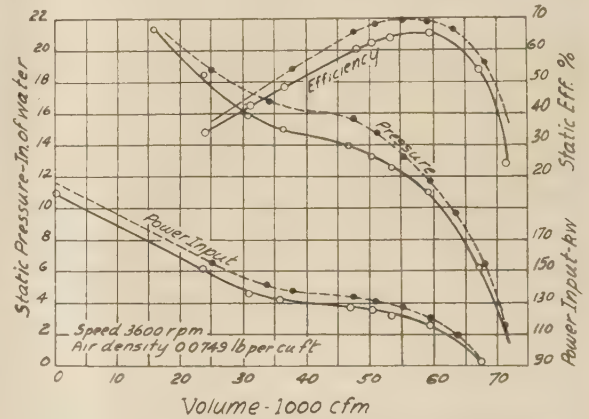

Fig. 6 Performance Curves of the Fan With Diffuser Only
(Performance with very close clearance between the housing and fan is shown by dashed lines while the performance with a clearance of 1.25 per cent of the fan housing is shown by solid lines.)

given point on the blade section increases until a point is reached where the angle of attack has reached a value corresponding to the maximum lift. Beyond this point the decrease in lift which takes place on a wing section is replaced in the propeller-fan blade by centrifugal action. Thus, at the point where the pressure characteristic flattens out, the airfoil theory no longer holds.

Tests With Diffuser and No Guide Vanes. The performance curves for the fan with diffuser and no guide vanes are shown in Fig. 6. A large gain in efficiency results when the diffuser alone is used. In this particular fan, a reasonably large gain using a diffuser alone may be expected because approximately 20 per cent of the fan power input is necessary to create the axial component of the dynamic energy present at the fan-discharge area. Of this 20 per cent, 3.2 per cent represents the axial component of the dynamic energy at the discharge area of the diffuser. The change in static efficiency of the fan from 56 to 66 per cent with the diffuser and same housing clearance in each case indicates that the diffuser recovered about 60 per cent of the available dynamic energy between the entrance and exit of the diffuser. Since no guide vanes were present to straighten the rotating air, the 80 per cent conversion of axial dynamic energy to static energy which is to be expected with a good diffuser could not be obtained. A test made with the center cone of the diffuser removed reduced the static efficiency from 66 per cent to 64 per cent.

Guide-Vane Model Tests. The proportions and angles of the guide vanes were determined by a set of tests. It was found that flat-plate vanes twisted so that the angle varied from the hub to tip gave but a small increase in static efficiency. The plate shape is much more sensitive to the direction at which the air leaving the fan strikes the vane than the blunt-nose airfoil-shaped vane. The air leaving the fan strikes the airfoil vane at a certain angle of attack. With the blunt-nose airfoil-shaped vane, the angle of attack may vary as much as 10 to 12 deg before the air flowing around the vane tends to "break" away from the vane

surface. At the "break" point the vane no longer acts as a guide to straighten the air flow from the rotational direction to the axial direction, but acts instead as an obstruction. The approximate angles at which the air was discharged from the fan were measured from hub to tip by means of a thread and protractor. From these

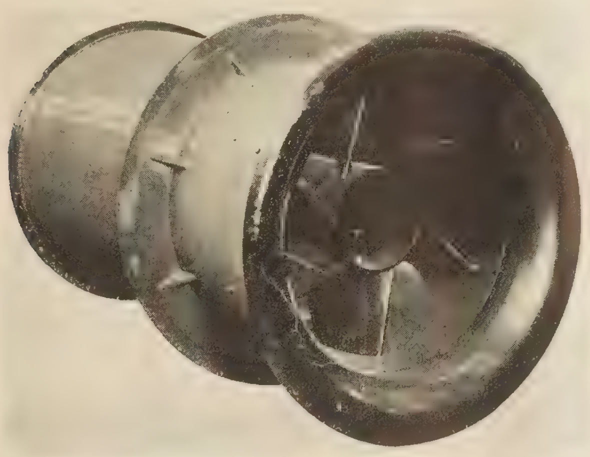

Fig. 7 The Diffuser and Guide Vanes of the Blower

data several sets of airfoil-shaped vanes with various angles of twist were made and tested. It was found that the vane which gave the largest increase in static efficiency near the desired fan working point was set within about 5 to 8 deg of the theoretical discharge angle corresponding to the calculated rotation of the fan.

The number of guide vanes was varied and it was found that when less than six vanes were used the gains in efficiency were reduced, probably due to the lack of sufficiently close spacing between vanes to provide the required guiding action. Both the six-vane and eight-vane arrangements gave about the same efficiency. The six-vane arrangement was used on the final full-size unit. The guide vanes mounted in the diffuser are shown in Fig. 7.

Full Size Tests and Results

Description of Test Set-Up and Method of Test. The full-size unit assembled with inlet box and diffuser is shown in Fig. 8. A large discharge box was built at the discharge of the diffuser. A portion of this box can be seen in Fig. 8. On the roof of the discharge box, five stacks, each with a different discharge area, were constructed and used as flow nozzles. Various volume points on the performance curve of the fan were obtained by closing different combinations of the stack openings.

The air enters the inlet box from above and passes through the fan and diffuser to the discharge box. A large screen with a cheese-cloth covering is located in the discharge box just beyond the diffuser outlet. This serves to break up the velocity head of the air leaving the diffuser and thus provides uniform inlet conditions to the nozzles.

The air volume was measured by making a pitot-tube traverse at the exit of each nozzle. In order to investigate the air distribution in each nozzle and obtain an accurate average, a large number of pitot-tube readings were taken. The air distribution was found to be reasonably uniform and the direction of the air was essentially parallel with the axis of the nozzle. These conditions made it possible to measure the air volume very accurately. The use of the large discharge box and screens eliminated any errors which might be present due to pulsating flow conditions.

The static pressure generated by the fan was measured in the discharge box after the air had passed through the screen. At this point the air velocity was very low and thus an accurate measurement of static pressure could be obtained.

The fan is driven by a 3600-rpm induction motor. The power output of the motor was accurately obtained by means of a special calorimeter-test method which will not be described here. This method of test eliminated the uncertainty of the magnitude of the load loss which in this case was enough to affect the fan efficiency by approximately 2 per cent.

Test Results. The results of the test on the full-size fan with guide vanes and diffuser are given in Fig. 9. The performance includes the effect of the inlet box. The power input to the fan includes the loss in the thrust bearing which is of the order of 1.5 kw. The model tests with similar guide vanes and diffuser checked the full-size performance closely. The maximum static efficiency on the model was 77 per cent. The maximum static

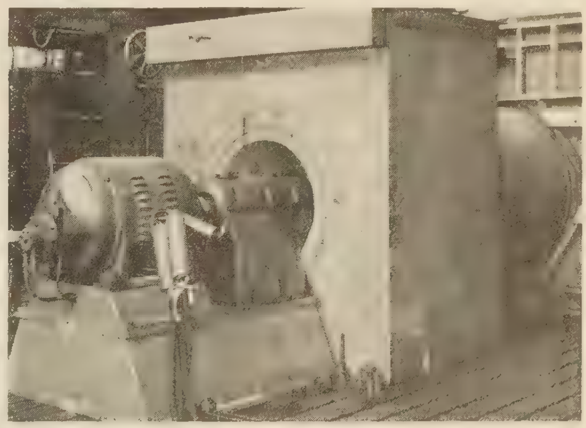

Fig. 8 Test Arrangement of the Assembled Full-Size Blower Unit

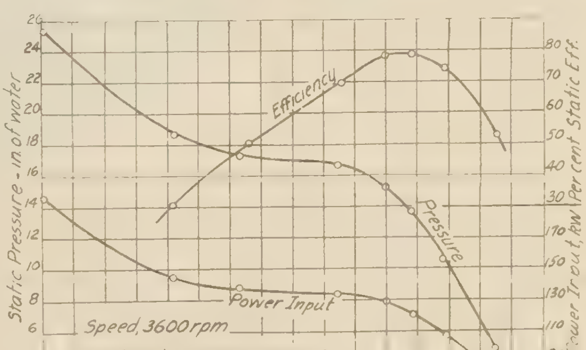

Fig. 9 Performance Curves of the Full-Size Unit With Guide Vanes and Diffuser

efficiency on the full-size unit was 78.5 per cent, including the loss due to the fan thrust bearing. The close check between the tests of the model and full-size unit indicated that the different methods of test used on the model and the full-size unit were satisfactory. The total efficiency of the unit with guide vanes and diffuser based on the static pressure plus the axial dynamic pressure at the diffuser discharge is 82.5 per cent, including the effect of the inlet-box and thrust-bearing loss.

Clearance Between Fan and Housing. The model tests demonstrated the importance of the clearance space between the outside fan diameter and the surrounding housing. Fig. 6 shows performances ascertained from two tests, one with a clearance between fan and housing of 1.25 per cent of fan radius, and the other with a very small clearance obtained on the model by molding a layer of modeling clay to the inside of the housing and scraping the excess off by running the fan in the housing. With this type of fan, an appreciable gain in efficiency results when clearances are close. The axial width of the eight-bladed fan at the blade tip is about 24 per cent of the radial height of the blade. The small axial width compared to the blade height may be linked with the sensitivity to clearance. The clearance between the blade tip and housing for the full-size unit is 0.163 per cent of the outside fan radius. In the test shown in Fig. 5, the efficiency of the fan itself was raised from 57 per cent to 61 per cent when an extremely small clearance was used.

Calculated Losses and Efficiencies Compared to Tests

The losses in a propeller fan may be classed under the two headings of recoverable and non-recoverable losses. The non-recoverable losses are those which originate in the fan itself and are called the drag losses. It is equivalent to the drag loss on an airfoil and is due to the fact that each blade element in producing a certain lift must overcome a certain drag due to friction and the nature of the flow around the airfoil.

The recoverable losses consist of the energy in the air at the discharge side of the fan. This energy is conveniently resolved into two components, namely, the rotational-velocity component which lies in a place perpendicular to the axis of rotation, and the axial-velocity component which is parallel with the axis of rotation of the fan.

Table 2 gives the magnitude of these various losses expressed as per cent of energy input to fan at working point of fan itself. (For point of maximum efficiency see Fig. 5.) Table 2 shows which losses limit the efficiency of the fan. The agreement between the calculated and test values of static efficiency indicates that the magnitude and relative apportionment of the total losses into their various parts is approximately correct.

A study of Table 2 makes the following points worthy of note.

TABLE 2 CALCULATED LOSSES AND EFFICIENCIES, PER CENT OF FAN INPUT

Item	Fan alone	Fan with guide vanes and diffuser
(*a*) Drag loss, not recoverable	13.3	13.3
(*b*) Rotational-loss component	9.3	3.7[a]
(*c*) Axial dynamic-energy component	20.0	3.4[b]
(*d*) Energy at diffuser exit, not recoverable	0	3.2
(*e*) Total losses, Items *a* + *d*, not recoverable	13.3	16.5
(*f*) Total losses, Items *b* + *c*, recoverable	29.3	7.1
(*g*) Total losses, Items *a* + *b* + *c* + *d*	42.6	23.6
Calculated static efficiency (100 — Item *g*)	57.4	76.4
Test static efficiency	56.5	78.5[c]

[a] Assumed 60 per cent recovered.
[b] Assumed 80 per cent recovered.
[c] Approximately 1 per cent low because of bearing loss.

If all of the recoverable losses were recovered, the maximum efficiency of the fan would be 86.7 per cent. The approximate percentage of losses recovered by means of the guide vanes and diffuser amounts to 75 per cent of the recoverable losses. The largest component of loss is the axial dynamic component which explains in part why the diffuser alone without the guide vanes gave a large increase in the static efficiency.

Noise

No complete investigations of noise levels were made but some comparisons indicated that the noise level of the fan would add little, if any, to the general noise level of the power station. These comparisons are based on the fan operating on the test floor where the intakes were open, but in the actual application the fan is part of a completely enclosed system.

Summary

The results of designing, building, and testing a fan of this type bring out the following possible advantages:

1 Within certain limits the performance and design proportions can be predetermined accurately.

2 From a mechanical standpoint, the propeller is particularly adaptable for high speeds.

3 The intake box has little effect on the fan performance.

4 For the pressures and volumes used in this application, the size of the unit is reasonably small.

5 The fan efficiency is high.

Progress Report on Cavitation Research at Massachusetts Institute of Technology

By J. C. HUNSAKER,[1] CAMBRIDGE, MASS.

This paper, a second progress report on the study of the nature of hydraulic cavitation being conducted at M.I.T., is continued along the lines reported in the first paper published in *Mechanical Engineering*, April, 1935, and discusses information obtained regarding the relation between frequency of vapor collapse and the controlling variables. The current work tends to confirm that previously reported. A new method for the study of resistance to cavitation attack is described.

IN THE present work on hydraulic cavitation it is found that the coefficient fL/V remains substantially constant, not only for geometrically similar venturi-type nozzles but also for all nozzles having the same exit flare. In the coefficient term fL/V, f is the frequency of collapse, L is the length of cavitation volume, and V is the velocity at the nozzle throat. Reynolds' number does not appear to control matters, possibly because the range for variation of Reynolds' number in the tests is not very great, but more probably because the flow is inherently turbulent and viscosity effect cannot be important. However, the relation fL/V can only be determined when cavitation is well developed and L is of substantial magnitude.

Further work has confirmed the opinion that the damage to surfaces caused by cavitation is severe only when the collapse is noisy, violent, and regular. For such fully developed cavitation the frequency of collapse is regular. When cavitation is not well developed and L is but one or two throat diameters, the frequency observed is highly irregular, severe shocks are infrequent or absent, and damage to walls is very slight.

Fig. 1 Lead Disk $^5/_8$ In. Diameter After 0.5-Min Oscillation in Calcium-Chloride Solution

Wall damage also appears to depend on the abolute size of the nozzle used, or the energy available in the cavitation volume, since tests with small nozzles, geometrically similar to large ones produced little or no damage even when cavitation conditions appeared to be similar.

When the exit flare varies more or less from 21 deg, the frequency of cavitation collapse is irregular, severe shocks infrequent or absent, and damage slight regardless of the length of cavitation volume for equal throat velocities.

The conclusion from the current work is that cavitation of a type to cause severe damage requires, probably, some unique combination of variables, of which exit flare or profile shape appears to be important. Therefore, a technique for the quantitative measure of the severity of cavitation damage is required. In this connection exposure of test medallions of metal has proved of interest. The determination of weight loss against time of exposure, however, does not give a suitable numerical measure of the severity in a reasonable time. Lead, for example, may be severely damaged without showing appreciable loss of weight. However, it is found that properly cured cement will progressively lose weight under cavitation attack and it is planned to report tests of various cavitation conditions in terms of a scale based on loss of weight of cement.

The severity of cavitation not only depends on the type of cavitation but must also depend on the liquid, and especially its density and vapor pressure. However, it is impractical to repeat the hydraulic experiments with other liquids.

It is known, however, that high-frequency vibration of a submerged diaphram, such as is used in submarine signaling, can produce cavitation and damage. An apparatus, similar to that described by Gaines[2] for oscillating a nickel tube at high frequency by magnetostriction effect was used to oscillate $^5/_8$-in. disks of lead and cement with a frequency of about 8500 cycles per sec through an amplitude of about 0.01 mm.

Tests are being made with a number of common liquids of widely different physical properties. Tests of lead disks in water or in a calcium-chloride solution show vastly more damage than when tested in alcohol or in carbon tetrachloride. Presumably the high vapor pressure of the latter liquids accounts for this. The appearance of the damaged surface of the lead is identical

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

[2] "A Magnetostriction Oscillator Producing Intense Audible Sound and Some Effects Obtained," by N. Gaines, *Physics*, vol. 3, no. 5, 1932, pp. 209–229.

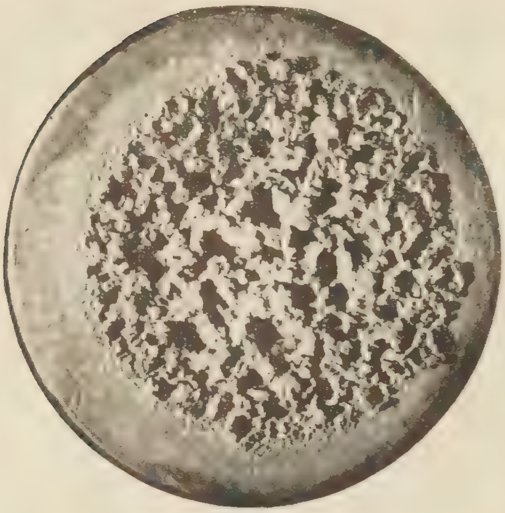

FIG. 2 LEAD DISK 5/8 IN. DIAMETER AFTER 6 MIN OSCILLATION IN CALCIUM-CHLORIDE SOLUTION

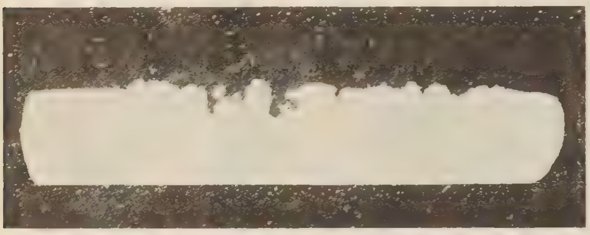

FIG. 3 CROSS-SECTION OF THE DISK SHOWN IN FIG. 2

with that observed on lead exposed to hydraulic cavitation in the original apparatus. However, the damage to lead is not consistently measured by loss of weight and the tests are being repeated with cement disks. The results of these tests will be reported later.

It is found that either with lead or with cement, appreciable damage is done in one minute and a test can be completed in less than 10 minutes. Fig. 1 shows the surface of a lead disk after oscillation of one-half minute duration in saturated calcium-chloride solution. Figs. 2 and 3, the latter a cross-section of the disk in Fig. 2, show a disk similar to the one in Fig. 1 after six-minutes exposure. It will be noted that material on the surface is displaced plastically and not lost. On the other hand, a cement surface is not displaced plastically and damaged particles are washed away. Fig. 4 shows the appearance of a cement disk after 1.2 min exposure to oscillation in water.

FIG. 4 CEMENT DISK 5/8 IN. DIAMETER AFTER 1.2 MIN OSCILLATION IN WATER

This new method of high-frequency oscillation permits a rapid estimation of the relative resistance to cavitation damage of various materials of construction. Likewise the method lends itself to a fundamental study of the influence of the properties of the liquid on cavitation damage. To oscillate a sample in a beaker of fluid for a few minutes is much cheaper and quicker than to run hydraulic cavitation tests.

Progress in Cavitation Research at Princeton University

By LEWIS F. MOODY[1] and ALFRED E. SORENSON,[2] PRINCETON, N. J.

The authors describe test apparatus and an experimental method used to determine the critical point of cavitation, or cavitation limit, of liquids flowing in a closed system as caused by the formation of vapor-filled cavities in the system. The experiments here described were directed to determine the cavitation limit, at which cavitation begins, and not to ascertain the consequences of maintained cavitation. The authors show that the experiments give results entirely consistent with the cavitation principle as now generally applied.

The Problem of Determining the Cavitation Limit

As THE effective head causing flow through a hydraulic apparatus is increased, the velocity head at any point in the system increases correspondingly and this increase in velocity head entails a compensating reduction in pressure head. This relation holds until the absolute pressure at some point in the system reaches the vapor pressure of the liquid, below which it cannot go while liquid remains present. Any further increase in the effective head merely produces boiling, the conversion of part of the liquid into vapor. When this occurs the principle of continuity of flow then fails to apply and a marked change in the performance of the apparatus occurs. The production of vapor-filled cavities in the flowing liquid is what is meant by the term *cavitation.* The consequences are an interconnected pressure pulsation and condensation and reevaporation of the vapor pockets constituting a series of explosions, or more strictly implosions, with resulting mechanical vibration and pitting of the conduit walls. The experiments here described were directed to determine the critical point or cavitation limit, at which cavitation begins, and not to ascertain the consequences of maintained cavitation.

The principles governing the incidence of cavitation in hydraulic machines such as turbines and pumps were analyzed and formulated by one of the authors at the Hydroelectric Conferences of 1922[3] and 1925[4] in Philadelphia and by D. Thoma at the First World Power Conference in 1924 at London,[5] and need not be repeated here. The use of Dr. Thoma's simple and convenient expression for the criterion of cavitation has become well standardized and will be adopted here. The relation is $\sigma = (H_b - H_s)/H$, in which H is the effective head producing flow through the system; H_b is the height of a water barometer or the atmospheric head minus the vapor-pressure head, that is, $H_b = H_a - H_{vp}$; H_s is the elevation of the point where cavitation begins above the free-water surface under static conditions such as the tail-water surface in the case of a turbine or pump; and σ is a coefficient which is a constant for a given apparatus. The relation is not limited in application to machines having moving parts, but is equally applicable to stationary conduits.

Questions have been raised as to the validity of this relation, and it has been stated by J. C. Hunsaker[6] in reporting on experiments on a special profile that "Experience with the profile shows that cavitation commences at pressures much higher than the true vapor pressure." (The authors suggest that the pressure here referred to may not have been that at the point of lowest pressure; also, that separation of air may be distinguished from true cavitation due to separation of condensable vapor.) It must be admitted that although the derivation appears to be based on logical deduction and its use in turbine practice seems to be satisfactory, nevertheless it is desirable to obtain direct experimental evidence of its validity or non-validity.

In the experiments here described, the formula was applied to a stationary conduit of simple form, namely a small venturi meter, in which the throat section can be relied upon to give good velocity distribution without curvature of the flow lines and to provide a piezometer location which may reasonably be expected to record the lowest pressure in the system. To avoid a varying elevation of the region of cavitation the conduit was kept horizontal. The entrance and discharge diameters were the same, so that the effective head producing flow was merely the difference in pressure heads, requiring no velocity-head corrections.

The Method Applied

As an indication of the performance of the apparatus, the authors used a ratio of heads representing the efficiency of conversion of velocity head into pressure head in the diffuser, or enlarging tapered section; that is

$$e = (h_{p3} - h_{p2})/(h_{p1} - h_{p2})$$

in which h_{p1}, h_{p2}, and h_{p3} are the pressure heads at entrance,

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[2] Assistant Professor of Engineering, Princeton University, Princeton, N. J. Jun. A.S.M.E. Professor Sorenson was graduated from Rensselaer Polytechnic Institute in 1924 with an M.E. degree and from that year until 1926 was engaged as instructor of mechanical engineering at the school from which he was graduated. Since 1926 he has been employed in his present position.

Contributed by the Hydraulic Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers to be held in New York, N. Y., December 2 to 6, 1935.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

[3] "The Hydraulic Turbine in Evolution," by H. B. Taylor and L. F. Moody, *Engineers and Engineering*, Engineers Club of Philadelphia, vol. 39, July, 1922, pp. 241–259.

[4] "Inter-Relation of Operation and Design of Hydraulic Turbines," by F. H. Rogers and L. F. Moody, *Engineers and Engineering*, Engineers Club of Philadelphia, vol. 42, July, 1925, pp. 169–187.

[5] "Experimental Research in the Field of Water Power," by D. Thoma, Transactions, First World Power Conference, London, 1924, vol. 2, pp. 536–551.

[6] "Cavitation Research at M.I.T.," by J. C. Hunsaker, *Mechanical Engineering*, vol. 57, April, 1935, pp. 211–216.

throat, and discharge, respectively, expressed as heights of water column above the centerline of the conduit, and e is the efficiency of the diffuser based on pressure heads (without introducing the small velocity heads at entrance and discharge). This ratio was used as an indicator because the deceleration process may be expected to be particularly sensitive to change of flow pattern. The tests were run to determine whether the point of cavitation as evidenced by a sudden dropping-off in efficiency or discontinuity in the efficiency curve would coincide with the reaching of the vapor pressure at the throat of the conduit. The difference in pressure heads between discharge and throat is dependent on the velocity head and may be expressed as a coefficient σ multiplied by the effective head producing flow or

$$h_{p3} - h_{p2} = \sigma H$$

The absolute pressure at the throat is

$$H_{p2} = h_{p2} + H_a = h_{p3} + H_a - \sigma H$$

in which H_a is the atmospheric pressure head. As H is increased, H_{p2} decreases until it reaches the limit H_{vp}, the head corresponding to the vapor pressure. This according to the theory is the cavitation point, and the corresponding σ can be found from the equation

$$\sigma = (h_{p3} + H_a - H_{vp})/H = (H_b + h_{p3})/H$$

in which H_b is the barometric head $H_a - H_{vp}$.

If the pressure at discharge is lower than atmospheric, and the top of the water column is H_s ft below the elevation of the throat, then h_{p3} becomes $-H_s$, and $\sigma = (H_b - H_s)/H$, the form used for turbines or pumps.

By plotting the efficiency of the diffuser e versus σ, a point of discontinuity in the curve will indicate a change in the mode of flow such as would be caused by the formation of a vapor pocket. If the theory is correct this point should coincide with the point where the absolute throat pressure just reaches the vapor pressure of the water.

Finally, the results were plotted, as shown in Figs. 4 and 5, from which it will be seen that the break in the curve of efficiency versus σ, Fig. 5, coincides exactly with the reaching of the vapor pressure at throat as shown by Fig. 4. The corresponding values of the throat pressure are also shown in Fig. 5 for various values of σ, to show this coincidence more clearly. The slight slope of the efficiency curves in the region of no cavitation reflects the changing Reynolds number. The two curves for different ranges of back-pressure show agreement with the theory in each case.

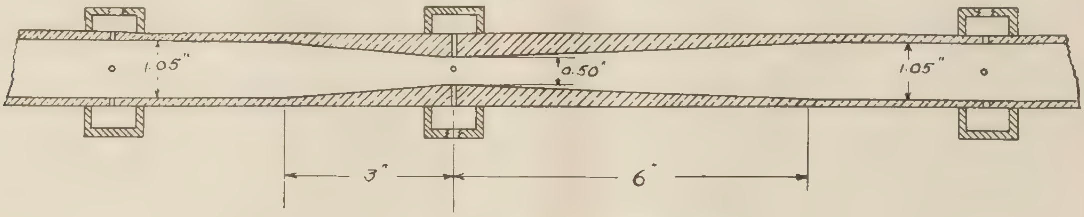

Fig. 1 Cross-Section of Venturi Meter

Apparatus and Procedure

The work described in this paper was conducted in the hydraulic laboratory of the Princeton University School of Engineering with the apparatus layout shown in Figs. 1 and 2. The venturi meter used was of brass, manufactured in the shop at the University. Before the tests were made, the instrument was carefully cleaned and the piezometer openings smoothed off with fine emery cloth. The throat diameter was 0.50 in. and the upstream and downstream sections measured 1.05 in. in diameter.

The water for these experiments was pumped from the laboratory sump to a tank on the top floor of the building from which it flowed by gravity through $1^1/_2$-in. piping to the venturi meter. The water was then discharged through $1^1/_2$-in. piping into a pair of weighing tanks which were not used in this series of experiments.

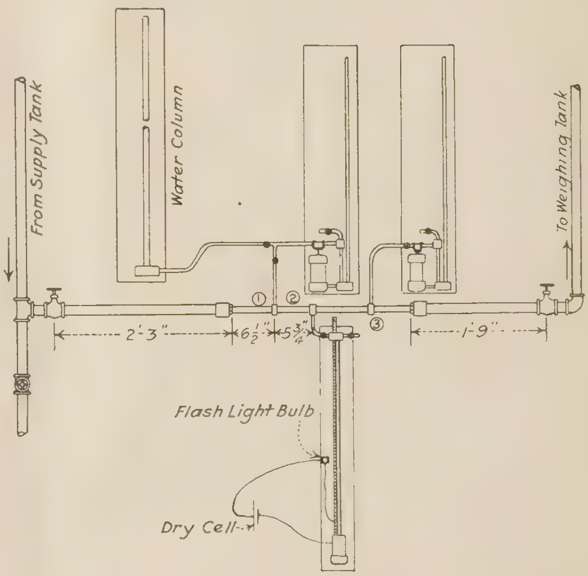

Fig. 2 General Layout of Apparatus

Many types of gages were used before suitable instruments were obtained. There was so much oscillation in the plain U-tube manometer that it had to be modified for use in this connection. The gages for measuring the pressures at points 1 and 3 in Fig. 2 were made as shown by using a 2-in. pipe nipple capped on both ends, as a mercury pot. The pot was connected by a $^1/_8$-in. nipple to a cast-iron header which held the gage glasses. The gage glasses were $^3/_8$-in. outside diameter and $^1/_4$-in. inside diameter and were held in the header by means of rubber stoppers. Meter sticks graduated in millimeters were used as scales on all gages. The gage for recording the pressure at the throat of the meter had to be made differently because of the vacuum at this point. The gage which was finally successful was similar to a barometer but the scale was adjustable and was fitted with a strip of stainless steel set at its exact zero mark. The strip was connected through a dry cell to a flashlight bulb, so that when the point touched the mercury level in the pot, the circuit was completed and the bulb would light. It was found that this arrangement was very sensitive and gave excellent results.

The top of the gage glass was connected by means of a cast-iron header to the throat of the venturi. This header had to be made extremely tight and was boiled in paraffin to close all pores. As finally used, this gage was able to hold a 28-in. vacuum overnight. Connections to all gages were made with $^1/_4$-in. copper tubing and Dole compression fittings. Valves and vent valves on the gages were $^1/_8$-in. Dole needle valves. A water column 10 ft high made of $^1/_2$-in. glass tubing was used as a calibrating device for the gages.

The specific gravity of the mercury was first checked by finding the deflection of the mercury manometers as compared to a given height of water column. This varied slightly from day to day due to changes in temperature.

The downstream valve was opened up as wide as possible and the flow regulated by the valve upstream. It was found that as the quantity of water passing through the meter increased, the absolute pressure at the throat of the meter decreased to a certain point when it became constant no matter how much the velocity of the water at the throat was increased. This then should be the vapor pressure of the water. However, this pressure was always between 2 and 3 in. of water lower than the vapor pressure shown by the steam tables for water at the temperature used in the tests.

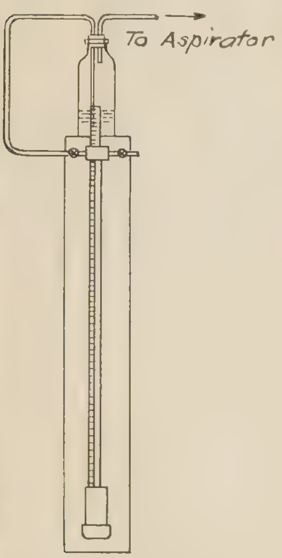

Fig. 3 Apparatus for Finding Vapor Pressure

The water used in the tests was obtained from a well and it was found to be fairly hard. It was decided then to find the vapor pressure of the water. This was done by means of a static water column connected to a bottle and connected in turn to an aspirator, as shown in Fig. 3, forming a water barometer. This procedure gave a vapor pressure for the water which corresponded exactly with the minimum pressure found in the venturi meter. In order to check this method of determining vapor pressure, distilled water was placed in the bottle and gage connections, and the experiment repeated. The vapor pressure found in this way agreed with the value given in the steam tables for water at the temperature which was being used.

A series of runs was made with the discharge valve wide open and a second series with this valve partly throttled. For each series of runs, the pressure at the throat of the meter was progressively lowered by adjusting the inlet valve, runs being taken at suitable intervals. As many points as possible were taken just before cavitation took place and also just after it had occurred. Also a number of runs were taken beyond the cavitation point. For each run, the pressures at the entrance to the meter, the throat of the venturi, and the exit from the instrument were read as nearly as possible at the same time. Since cavitation should begin at the upper wall of the throat rather than at the centerline, the heads were corrected to correspond to this point as a datum elevation. The observed data taken during the runs and the results of the tests are given in the appendix of the paper. Before the meter was cleaned out, no noticeable noise was produced when cavitation occurred, but after the greasy deposit on the walls had been removed, cavitation caused a continual noise with periodic major disturbances causing a considerable variation in pressure at points 1 and 3, Fig. 2. This sound effect gives secondary evidence of the beginning of cavitation.

Conclusions

While the authors have no basis for asserting that the cavitation formula will satisfy all conceivable arrangements and conditions, these experiments appear to answer the question of its validity, for the apparatus and conditions adopted, by confirming it. This was true for two different values of back pressure, showing that σ remains constant and independent of the back pressure, in accordance with the theory. It was found that when dealing with hard waters, heavily charged with salts, the vapor pressure may be appreciably higher than that given for distilled water in the steam tables, and when applying the formula to unknown waters a small allowance should be provided in the value of H_b to cover this factor. At least when using ordinary run-of-laboratory water free from abnormal air content, the formula appears to be perfectly satisfactory, without modification.

In brief, these experiments give results entirely consistent with the cavitation principle as now generally applied.

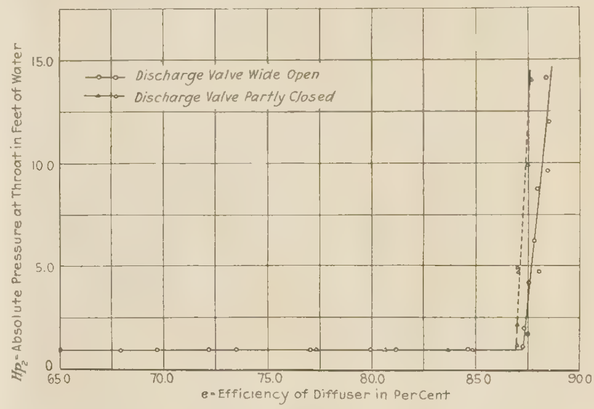

Fig. 4 Curve of Absolute Pressure at Throat of Meter Vs Efficiency of Diffuser (H_{p2} Vs e)

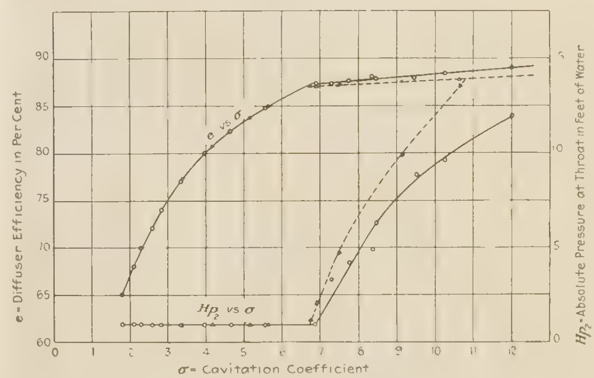

Fig. 5 Curves of Efficiency of Diffuser Vs σ (e Vs σ) Also Absolute Pressure at Throat of Meter Vs σ (H_{p2} Vs σ)

Appendix

Observed Data Taken to Determine Cavitation Limits

Run No.	Gage No. 1 Col 1	Gage No. 1 Col 2	Gage No. 1 Net	Gage No. 2 Hg col	Gage No. 2 Hg ht below datum	Gage No. 3 Col 1	Gage No. 3 Col 2	Gage No. 3 Net
1	3.00	10.64	7.64	40.48	134.1	3.34	5.06	1.72
2	2.66	11.42	8.76	45.66	134.0	3.70	6.20	2.50
3	2.94	14.20	11.26	51.46	134.2	3.32	7.10	3.78
4	2.76	14.36	11.60	53.54	134.2	3.80	7.32	3.52
5	2.88	18.04	15.60	59.56	134.4	3.36	9.18	5.82
6	2.80	18.54	15.74	62.12	134.3	3.80	9.36	5.56
7	2.52	18.74	16.22	64.66	134.4	3.60	9.64	6.04
8	2.80	21.58	18.78	66.58	134.5	3.32	11.04	7.72
9	2.72	21.14	18.42	69.32	134.4	3.64	10.86	7.22
10	2.40	20.66	18.26	70.64	134.5	3.54	10.66	7.12
11	2.74	25.00	22 26	72 10	134.6	3.20	13.40	10.20
12	2.60	30.68	28.08	72.50	134.6	3.20	15.94	12.74
13	2.20	30.56	28.36	72.40	134.6	3.48	13.96	10.48
14	2.50	34.72	32.22	72.45	134.6	3.12	14.18	11.06
15	2.40	39.52	37.12	72 40	134 6	3.20	15.06	11.86
16	2.30	44.96	42.66	72.40	134.6	3.20	15.82	12 62
17	2.30	48.92	46.62	72.40	134.6	3.18	16.74	13.56
18	2.20	54.80	52.60	72.40	134.6	3.20	17.72	14.52
19	2.10	59.06	56.96	72.40	134.6	3.22	18.46	15.24
20	1.96	66.30	64.34	72.40	134.6	3.16	19.42	16.26
				Second Series—Discharge Valve Throttled				
1	2.90	22.10	19.20	42.00	134.1	3.50	15.04	11.54
2	2.76	27.00	24.24	50.62	134.2	3.44	18.34	14.90
3	2.16	27.88	25.72	62.86	134.3	3.44	17.52	14.08
4	2.56	34.88	32.32	63.34	134.4	3.36	23.48	20.12
5	2.50	40.32	37.82	68.03	134.4	3.30	26.92	23.62
6	2.14	31.56	29.42	69.42	134.5	3.38	19.88	16.50
7	2.46	42.26	39.80	71.20	134.9	3.30	28.68	25.38
8	2.36	45.88	43.52	72.40	134.6	3.30	29.40	26.10
9	2.30	49.50	47.20	72.40	134.6	3.30	31.18	27.88
10	2.10	59.82	57.72	72.40	134.6	3.20	35.90	32.70
11	2.00	67.78	65.78	72.40	134.6	3.18	37.74	34.56

Results of Tests to Determine Cavitation Limits

Run No.	H_{p1} ft gage	H_{p2} ft gage	H_{p2} ft abs	H_{p3} ft gage	$\frac{H_{p3} - H_{p2}}{H_{p1} - H_{p2}}$	$\frac{H_b - H_v}{H}$
1	3.122	—19.570	14.123	0.484	0.884	12.590
2	3.643	—21.716	11.993	0.843	0.890	12.000
3	4.730	—24.097	9.596	1.399	0.884	10.248
4	4.910	—24.971	8.738	1.299	0.879	9.431
5	6.460	—27.440	6.253	2.308	0.878	8.440
6	6.734	—28.520	4.970	2.534	0.881	8.355
7	6.959	—29.562	4.147	2.416	0.876	7.742
8	8.070	—30.335	3.358	3.152	0.872	7.297
9	7.925	—31.492	1.998	2.916	0.873	7.082
10	7.863	—32.031	1.678	2.894	0.875	7.174
11	9.616	—32.615	1.078	4.251	0.873	6.894
12	12.199	—32.777	0.916	5.336	0.847	5.547
13	12.351	—32.756	0.953	4.388	0.823	4.665
14	14.037	—32.756	0.937	4.630	0.799	3.972
15	16.213	—32.736	0.957	4.989	0.771	3.361
16	18.674	—32.736	0.957	5.327	0.740	2.852
17	20.435	—32.736	0.957	5.701	0.723	2.608
18	23.092	−32.736	0.957	6.190	0.697	2.303
19	25.027	—32.736	0.957	6.493	0.679	2.117
20	28.305	—32.736	0.957	6.945	0.650	1.858
			Second Series—Discharge Valve Partly Closed			
1	8.277	—20.196	13.497	4.770	0.877	10.695
2	10.515	—23.750	9.943	6.242	0.875	9.122
3	11.174	—28.800	4.909	5.989	0.870	7.473
4	14.102	—28.997	4.696	8.561	0.871	7.453
5	16.554	—30.981	2.509	10.207	0.866	6.738
6	12.821	—31.528	2.181	7.064	0.870	6.917
7	17.435	—32.268	1.222	10.990	0.870	6.757
8	19.077	—32.736	0.957	11.219	0.848	5.594
9	20.711	—32.736	0.957	12.010	0.837	5.143
10	25.383	—32.736	0.957	14.151	0.807	4.174
11	28.965	—32.736	0.957	14.977	0.773	3.411

Notes on First Table

Datum plane level = 45.87 cm above center line of venturi meter; gage 1 datum level = 61.35 cm; gage 2 datum level = variable; gage 3 datum level = 61.58 cm; all readings in this table are average values; all readings are given in centimeters.

The Measurement of Air Flow in Fan Inlet and Discharge Ducts

By LIONEL S. MARKS,[1] CAMBRIDGE, MASS.

This paper is devoted to a proposed method for determining the volume of air handled by any fan which is provided with either inlet or discharge ducts. The proposal is to use pitot tubes for capacities near the wide-open condition and square-edged inlet or discharge orifices for lower capacities down to the no-flow condition. To make the pitot-tube measurements accurate, a standard form of pitot tube is suggested, based on the investigations of Ower[2] and the unpublished researches of Merriam and Spaulding at Worcester Polytechnic Institute. The coefficients of Ebaugh and Whitfield[3] are suggested for use with inlet orifices, while those of Stach[4] are suggested for discharge orifices used with flange pressure taps on the approach side. The alternative procedure of using an impact tube with the discharge orifice is discussed, but coefficients are not as yet available.

IN MAKING tests of fans with inlet or discharge ducts, the volume of air to be measured will vary from a maximum to zero, the variation being obtained by changing restrictions on either the inlet or the discharge side of the fan. The restriction may either be separate from the volume-measuring device or may be an essential part of it.

The volume-measuring devices and restrictions which come under consideration are (1) the pitot tube, with independent restrictions; (2) a single nozzle in the duct (duct nozzle) with independent restrictions, or a series of nozzles used individually, acting as restrictions; (3) a series of inlet or discharge nozzles at the end of the duct, acting as restrictions; (4) an orifice in the duct (duct orifice) with independent restrictions, or a series of orifices acting as restrictions; and (5) a series of inlet or discharge orifices, acting also as restrictions.

No one of these devices is satisfactory for the complete range of operation of a fan. The pitot tube will not give accurate indications at low fan capacities; nozzles and orifices will not permit the investigation of a fan with wide-open discharge. When a large single nozzle or orifice is used (with supplementary restricting devices), it will not give accurate indications at low capacities; while if the single nozzle or orifice is small, it will not permit fan operation near the wide-open condition.

As between nozzles and orifices, assuming that the coefficients of both are equally definite for stated conditions and that they have been determined with equal accuracy, the choice must fall on the orifice. In fan testing, the ducts may be as large as 10 ft diameter. Well-rounded nozzles of large size are costly and heavy. Other nozzle forms, fabricated from sheet metal, have been proposed, but they cannot be made inexpensively of any desired exact form and dimensions, and consequently their coefficients cannot be ascertained with accuracy. Moreover, they are heavier and more costly than orifices. The one advantage of a well-rounded nozzle is that its coefficient can be determined with sufficient accuracy by means of a traverse with a small impact tube. Such a nozzle is more nearly a primary standard for the measurement of large quantities of air than any of the other devices proposed. As such it may be used for the calibration of other measuring devices such as the orifice. When the coefficient of an orifice has been determined by the use of a nozzle or other means, it becomes just as accurate as the nozzle and is much cheaper and more convenient.

A duct orifice is practicable only for a single orifice because of the difficulty involved, and the time consumed, in changing orifices in a large-sized duct. A single orifice presents the disadvantages stated previously of having either low accuracy at low capacities or of preventing fan operation near the wide-open condition. Moreover, this method requires a considerable total length of duct. The only advantage of duct orifices over discharge orifices is that the coefficients for duct orifices have been well established by a number of investigators.

There remains the use of a series of orifices either at the entrance to the inlet duct (inlet orifices) or at the end of the discharge duct (discharge orifices). This is the simplest and most convenient method of measuring air and is comparatively inexpensive. Its only disadvantage is that it is not applicable to measurements at or near the wide-open condition.

It would appear from this survey that two of the methods enumerated for measuring the flow of air in a fan duct have definite advantages over all the other methods. The pitot-tube method can be made quite satisfactory at higher fan discharges, while the square-edged inlet or discharge orifice is equally satisfactory at lower fan discharges. A combination of the two methods should permit accurate volume measurements throughout the whole range of fan operation. The author proposes, therefore, that in testing fans which are to be used with inlet or discharge ducts, the air measurements be made by a pitot-tube traverse at capacities approximately down to the capacity at maximum efficiency and from there down, by square-edged orifices. Pitot-tube traverses should also be made while the two largest orifices are

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[2] "The Measurement of Air Flow," by E. Ower, Chapman and Hall, London, 1933; also, "The Design of Pitot-Static Tubes," by E. Ower and F. C. Johansen, Reports and Memoranda No. 981, Technical Reports of the Aeronautical Research Committee, August, 1925, London, England.

[3] "The Intake Orifice and a Proposed Method of Testing Exhaust Fans," by N. C. Ebaugh and R. Whitfield, Trans. A.S.M.E., vol. 56, December, 1934, paper PTC-56-3, pp. 903–912.

[4] "Die Beiwerte von Normdüsen und Normblenden im Einlauf und Auslauf," by E. Stach, *Zeitschrift des Vereines deutscher Ingenieure,* vol. 78, no. 6, February 10, 1934, pp. 187–189.

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in use so as to determine the accordance of the two methods of measurement.

Before the proposed method can be used satisfactorily, it is necessary (1) to standardize the pitot tube and methods of using it, and to know its errors, and (2) to standardize the square-edged orifice and methods of using it, and to ascertain its coefficients under various conditions of use.

Standardization of the Pitot Tube

The desirable characteristics of a standardized pitot tube are (1) known coefficients, eliminating the necessity for calibration; (2) ease of duplication with ordinary machine-shop processes; (3) insensitivity to minor changes in form and dimensions; (4) static holes as near the tip as possible; (5) rigidity; and (6) small size.

The features to be considered while designing such a standardized pitot tube are (1) diameters, thicknesses, and materials of the outer and inner tubes; (2) the shape of the impact tube; (3) the size of the impact opening; (4) the distance of the static holes from the tip; (5) the distance of the static holes from the stem; and (6) the size, number, and arrangement of the static holes.

The researches of Ower and Johansen,[2] and the unpublished researches of Merriam and Spaulding at Worcester Polytechnic Institute give the necessary data for selecting a design. The two entirely independent investigations, covering to a large degree the same field, give results remarkably accordant and justify complete confidence in their reliability.

The selection of diameters and thicknesses of the inner and outer tubes is arbitary. Adequate stiffness can be obtained for traversing ducts 3 or 4 ft across by making the outer tube of 0.04-in. thick brass (No. 18 gage), $^5/_{16}$ in. outside diameter, and the inner tube preferably of 0.0285-in. thick copper (no. 21 gage), $^1/_8$ in. outside diameter. Other sizes may be used without affecting accuracy as long as geometrical similarity is maintained.

The shape of the impact tube should preferably be hemispherical. A conical tip of proper design will give equally satisfactory results, but it is more liable to damage, increases the total length of the tube, and locates the static openings at a greater distance from the impact opening.

The diameter of the impact opening is immaterial when the tube is directed accurately into the air stream, but has considerable influence on the impact readings under conditions of yaw. This is illustrated, for a hemispherical head, by the following tabulation taken from the work of Merriam and Spaulding at Worcester Polytechnic Institute:

Ratio of impact opening to tube diameter	0.5	0.4	0.3	0.2
Error in impact reading, per cent of velocity head, with 16-deg yaw	3.3	4.5	6.0	8.5

The velocity-head reading under conditions of yaw will vary with both the impact reading and the static reading. If the errors of both impact and static readings in yaw were of the same sign and magnitude, the errors would cancel and the velocity-head reading would be correct. This desirable condition cannot be realized by any known design of pitot tube. With an impact opening 0.2 times the diameter and with the static-hole design recommended later in this paper, the velocity-head readings vary less with yaw than when the impact openings are larger.

If it is desired to find only the velocity or volume of the air flowing, an impact opening of 0.2 times the tube diameter should be selected. But in fan testing it is also necessary to find the work done on the air and this is proportional to the product of (1) the volume of air delivered, and (2) the impact pressure. An impact opening 0.2 times the tube diameter gives satisfactory values of the difference between impact and static pressures up to 12-deg yaw but gives larger errors in the values of the impact pressure, and of the work done on the air, than result from the use of an impact opening 0.4 times the tube diameter. This latter dimension appears to be the best to select for fan testing. With an effective straightener to eliminate yaw, all sizes of impact opening would give the same indications. The depth of this opening has no influence on the impact reading.

The variation of velocity head with yaw, and the error in the calculated work done on the air, with zero static pressure, for an impact opening 0.4 times the tube diameter (with stem-effect error included) is found to be:

Yaw angle, deg	8.0	12.0	16.0	20.0
Error in velocity head, per cent	—0.8	—1.7	—0.5	1.7
Error in work done, per cent	0.1	0.5	4.3	8.2

The error in calculated work done on the air diminishes as the static pressure is increased, being halved when the static pressure is equal to the velocity head.

The distance from the static holes to the tip of the tube, is made up of the length of the tip proper (conical or hemispherical portion) and a length of straight tubing between the tip proper and the static holes. The latter alone is found to have an influence on the static readings. This influence is considerable. The effect of the tip is to make the static reading less than the true static reading by an amount which depends on the length of straight tubing between the static holes and the tip, and on the velocity head. The error diminishes rapidly as the length of straight tubing increases, and becomes practically constant when this length is eight tube diameters. At this location the static-pressure reading is low by about 0.2 per cent of the velocity head according to Ower;[2] it is still smaller according to Merriam and Spaulding. This distance of eight tube diameters is indicated as desirable for the location of the static holes.

The distance from the static holes to the stem has an important influence on the static reading. According to Ower,[2] the presence of the stem increases the upstream static pressure as in the following tabulation:

Distance of stem aft of static holes, in tube diameters	5	10	15
Excess pressure due to stem, in per cent of velocity head	1.75	0.8	0.5

The investigations of Merriam and Spaulding show a slightly greater pressure excess. As this error cannot be eliminated without removing the stem to an impracticably great distance, and as the error changes very slowly after a length of 16 diameters, it seems desirable to select this length of 16 diameters for the distance between the static holes and the stem axis.

With the stem extending across the duct (which may be desirable for very large ducts) the static error has twice the value just given in the tabulation and, therefore, for a 16-diameter length, may be as much as 1 per cent of the velocity head.

The number of static holes and their distribution around the tube is determined principally by the consideration that the degree and direction of yaw, for the air current being measured, will ordinarily be unknown. Consequently the static holes should be uniformly distributed around the tube and should be numerous. The logical development of this concept leads to a gap instead of a number of holes, as in the the Prandtl tube. The objection to the gap is that it reduces the rigidity of the tube and in practice may lead to slight flexure at the gap. This would have the same effect as a burr and, while difficult to see, would materially affect the static reading. Eight holes with even spacing are suggested as a satisfactory compromise. If these holes are 0.04 in. diameter, they will take away less than one-third of the tube material.

The size of the static holes influences the static-pressure error

under yaw. Small holes (0.02 in. diameter) have the advantage of giving comparatively small errors in yaw but are likely to clog and yield static readings which under yaw conditions are very sensitive to changes in diameter. With holes 0.04 in. diameter, the following results were obtained by Merriam and Spaulding:

Yaw angle, deg	8.0	12.0	16.0	20.0
Static error, per cent of velocity head	1.5	3.5	5.5	8.0
Static error, including stem effect, per cent	1.0	3.0	5.0	7.5

It is proposed for the purpose of achieving comparative insensitiveness to slight imperfections, that the static holes be made 0.04 in. in diameter. One other feature is recommended. The inner tube should be maintained coaxial with the outer tube in the vicinity of the static holes. A spacing ring soldered to the

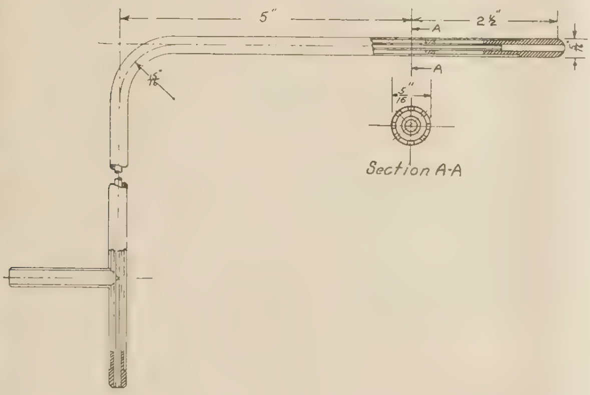

Fig. 1 Proposed Standard Pitot Tube for Fan Testing

copper tube will achieve this object. A pitot tube embodying the proposed features is shown in Fig. 1.

Square-Edged Orifices

The proposal to use square-edged orifices as a measuring device in fan testing is based upon the assumption that these orifices will always be comparatively large. If they are small, the condition of the edge of the orifice may influence appreciably the coefficient of discharge. Orifices above 8 in. diameter (approximately) will give predictable coefficients of discharge when they are machined with ordinary care and all burrs are removed. For smaller orifices, greater care is necessary; when the diameter is as small as 1 in., the orifice coefficient will be found to vary considerably with the condition of the edge of the orifice in which case the nozzle is preferable.

In addition to the minimization of the effect of edge sharpness when the orifice is large, there is the further fact that the coefficient varies but little with orifice size when the size is considerable. The elimination of these two disturbing factors leaves the orifice coefficient a function of two factors only: (1) the Reynolds number, and (2) the ratio of orifice area to duct cross-sectional area. These factors are considered later.

Inlet Orifices

Square-edged inlet orifices have been investigated recently by Ebaugh and Whitfield[3] for a 22³/₄-in. duct with a 20 to 100 per cent orifice-area ratio. Some earlier work by Watson and Schofield[5] deals only with small orifices from ¹/₂ to 2 in. diameter and for very small orifice-area ratios and is not applicable to fan conditions. Both these investigations show discharge coefficients very close to 0.6. The volume of air flowing is based entirely upon the pressure drop at the orifice and this necessitates a choice as to the location of the pressure tap in the duct. The Ebaugh and Whitfield coefficients[3] are based on the location of the pressure tap in the plane of the vena contracta.

The recent investigations of Stach,[4] using the I.S.A. form of square-edged orifice with flange taps and carried out on a 10-in. duct with orifice-area ratios of 0.25, 0.36, 0.49, and 0.64 give a constant value of the coefficient of discharge of 0.60 for Reynolds' numbers in excess of 55,000. The accuracy of the coefficient is given by the following percentage tolerances:

Orifice-area ratio	0.25	0.36	0.49	0.64
Tolerance, per cent	+ 1.3	± 1.0	± 0.5	± 0.5

It would appear, using flange taps, that the coefficient of discharge is not quite constant and that the assumption of constancy leads to larger errors than are permissible at low orifice-area ratios. As a practical matter, it is desirable to have a definite location for the downstream pressure tap. Ebaugh and Whitfield[3] find that a location 40 per cent of the pipe diameter downstream will permit the use of a constant coefficient of 0.601 throughout a considerable range of orifice-area ratios with an error of less than 0.5 per cent for ordinary pressure drops.

The large amount of research on pipe orifices cannot be considered applicable to the conditions of an inlet or a discharge orifice. The bibliography of Ebaugh and Whitfield[3] does not include any investigations which actually apply to an inlet orifice of a size suitable for the testing of fans. While their work needs verification by others and extension to larger ducts, greater pressure drops, and smaller orifice-area ratios, it may be accepted for the present.

Discharge Orifices

Square-edged discharge orifices have been investigated by Durley[6] up to 4¹/₂ in. diameter for very small orifice-area ratios. by Muller[7] up to 2¹/₂ in. diameter with large orifice-area ratios, and by Stach[4] for orifices from 5 to 8 in. diameter in a 10-in. duct.

The work of Durley[6] and Müller[7] does not cover the range of orifice sizes important in fan testing. The work of Stach[4] with I.S.A. orifices having flange pressure taps before the orifice, was for the same set of conditions as those stated previously, in the discussion of inlet orifices. The coefficients which Stach found are slightly lower than those for pipe orifices and vary with the orifice-area ratio m. The values are:

m	0.05	0.10	0.15	0.20	0.25	0.30	0.35
coefficient	0.598	0.602	0.608	0.615	0.624	0.636	0.651
m	0.40	0.45	0.50	0.55	0.60	0.65	0.70
coefficient	0.666	0.682	0.701	0.724	0.751	0.784	0.820

These values are found to hold for Reynolds' numbers R in excess of the following values:

m	0.25	0.36	0.49	0.64
R	45 000	55,000	75,000	183,000

The accuracy of these coefficients is given as ± 0.8 per cent. They apply to the simple flow equation $V^2 = 2gh$; and include the correction for velocity of approach.

The investigations of Stach[4] were made with a high degree of precision and his results apparently may be used with confidence.

The Reynolds numbers for inlet and discharge orifices in fan tests will ordinarily be in excess of the limit set by Stach.[4] The

[5] "On the Measurement of the Air Supply to Internal-Combustion Engines by Means of a Throttle Plate," by William Watson and H. Schofield, Proceedings of the Institution of Mechanical Engineers, May, 1912, pp. 517–552.

[6] "Measurement of Air Flowing Into the Atmosphere Through Circular Orifices in Thin Plates and Under Small Differences of Pressure," by R. J. Durley, Trans. A.S.M.E., vol. 27, 1906, pp. 193–231.

[7] "Messung von Gasmengen mit der Drosselscheibe," by A. O. Müller, *Zeitschrift des Vereines deutscher Ingenieure*, vol. 52, no. 8, February, 1908, pp. 285–290.

absolute viscosity μ of air varies from 1.175×10^{-5} at 32 F, to 1.51×10^{-5} at 212 F, in lb-ft-sec units, the variation being proportional to the temperature change. The Reynolds number is $DV\,\rho/\mu$, where D is the orifice diameter, ft; V is the velocity of air in the orifice fps; and ρ is the density of the air in lb per cu ft. At 70 F with standard air density of 0.075 lb per cu ft, $\rho/\mu = 6.05 \times 10^3$. Assuming the lowest practical pressure drop through the orifice as 0.1 in. of water, which corresponds to an air velocity of approximately 21 fps, and assuming an orifice diameter of 0.5 ft, the Reynolds number $R = 0.5 \times 21 \times 6.05 \times 10^3 = 6.35 \times 10^4$. The conditions here assumed are those which would give the minimum value of R that is likely to occur in fan testing. It will be seen that Reynolds' numbers for actual fan-test condition generally will be in excess of Stach's minimum for constant-discharge coefficients for square-edged orifices. Any doubtful case should be calculated.

The weight or volume of air flowing through a discharge orifice may be determined either from observations of the drop of static pressure through the orifice or from impact-tube readings. In recorded investigations, only the pressure-drop method appears to have been used. It would seem, however, that the impact-tube method is as applicable here as it is for nozzles and that it has the same advantages.[8]

A traverse along the center line of the orifice shows that for orifices 8 in. diameter or larger, the impact reading is constant from the plane of the orifice to several inches out from that plane, and that it falls off slowly at increasing distances. Therefore, it is proposed that the impact tube be placed at a distance of 1 in. out from the plane of the orifice.

The weight of air discharging from a square-edged orifice is given by the equation[9]

$$w = 0.1145\, c\, D^2 \sqrt{(GBi/T_1)}$$

where w is the weight of air flowing, lb per sec; c is the coefficient of discharge; D is the orifice diameter, in.; B is the mean discharge pressure, in. of mercury; i is the impact pressure, in. of water; T_1 is the temperature of the air approaching the orifice, deg F abs; and G is the specific gravity of the air referred to normal dry air at the same pressure and temperature. The quantity B is preferably the mean of the static pressures on the two sides of the orifice, since this gives a somewhat greater constancy of coefficient of discharge than the use of the discharge (barometric) pressure. For small pressure drops it is immaterial which of these pressures is used; with a pressure drop of 1 in. of water the difference in weight of air flowing is only 0.125 per cent, and for a pressure drop, of 4-in. of water it will be 0.5 per cent. No coefficients are available at present for the determination of the volume of air flowing through a square-edged discharge orifice by means of an impact tube.

A desirable form and method of assembly of the square-edged orifices is shown in Fig. 2. The orifices are made from $^1/_4$-in. steel plate and are provided with studs to facilitate assembly. The studs are round-headed machine screws, $^3/_8$ in. diameter, threaded the whole length, and with the thread cut away in the immediate vicinity of the head. These screws are threaded into holes spaced 6 in. apart in the orifice ring; the holes are slightly countersunk on the entering side. If screwed up tight, the screws will act as studs. The studs are $1^1/_2$ in. from the orifice and are found to have no perceptible influence on the flow. A ring of hard packing such as vellumoid is shellacked on the back of each ring and the exposed face of the packing is graphited. A series of these orifices can be built from any desired maximum size, by steps of at least 4 in., down to any desired minimum size.

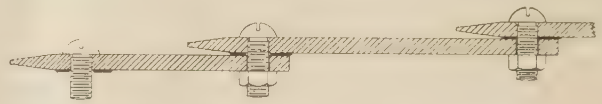

Fig. 2 Suggested Form and Method of Assembling Square-Edged Orifices

As shown in Fig. 2, the orifice plate is tapered on both sides for a distance of $^3/_4$ in. from the orifice, with a slope of 1 in. in 8 in., leaving a thickness of $^1/_{16}$ in. at the orifice. The purpose of this taper (which differs from the usual practice of having the approach side of the orifice flat and only the discharge side tapered) is to have an arrangement which will work equally well both for inlet orifices and discharge orifices. The justification is experimental. The author has investigated large orifices which were identical except that one was tapered, as in Fig. 2, and the other made in the usual manner with a flat approach side. The coefficients for orifices of this type are slightly higher than for the standard type.

[8] Discussion by S. A. Moss of "The Flow of Air and Steam Through Orifices," by H. B. Reynolds, Trans. A.S.M.E., vol. 38, 1916, p. 833.

[9] "Measurement of Flow of Air and Gas With Nozzles," by S. A. Moss, Trans. A.S.M.E., vol. 49–50, part 1, 1927–1928, paper APM-50-3.

The Definition of the Quantity-Head Characteristic of Fans

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The purpose of a fan characteristic is to delineate useful information regarding the performance of a fan in its installation. Specifically, the quantity-head characteristic shows the quantity of air delivered against a certain resistance in the intended installation. The authors discuss the set up of four basic arrangements of fan types and installations, and describe three currently used fan-head definitions. The definitions of fan head are evaluated, the fan head being defined as the discharge static head minus the inlet total head. A primary arrangement of each type of fan is described and a single primary quantity-head characteristic for each type of fan is defined. If a fan is installed in any arrangement other than its primary one, the primary characteristic cannot be assumed to apply, but a separate test must be conducted to determine its characteristic in its intended arrangement. Results of complete tests of a fan installed in its four possible arrangements are given to illustrate the principles which are set forth.

The purpose of a fan characteristic is to delineate useful information regarding the performance of a fan in its installation. Specifically, the quantity-head characteristic shows the quantity of air which will be delivered against a certain resistance in the intended installation.

In general, it may be stated that for a given fan there does not exist a single easily defined relationship between the quantity of air discharged and the head produced by the fan. The two circumstances accounting for this situation are (1) the lack of agreement upon a definition of the term "head" as applied to fan performance, and (2) the performance of a given fan is influenced markedly by the different arrangements required in various applications.

Description of Basic Fan Arrangements

An investigation of the different ways in which the arrangement of a fan in its application may affect the quantity-head characteristic reveals that four basic installation arrangements cover all cases. Before describing these basic arrangements the fan must be defined.

Prof. L. S. Marks in a series of papers[3] has shown that inlet boxes and other attachments at the inlets and outlets of fans have considerable effect on the performance of the fan in certain cases. Therefore, it is necessary in defining the fan to include not only the housing, but to have a definite understanding as to what attachments, such as inlet boxes, diffusers, mounting rings, flanges, dampers, etc., shall be considered as part of the fan. Having an understanding of what constitutes the fan, the four basic installation arrangements which are required to cover the various applications may now be developed. These basic installations are shown in Fig. 1 as arrangements A, B, C, and D.

Arrangement A—Bare Fan. In this arrangement there are no ducts attached to the fan; the intake is from one large space or room and the discharge is into another large space. This arrangement is that usually thought of in applications of propeller fans, although it is equally applicable to any housed fan. It may alternately be referred to as a wall installation.

Arrangement B—Blower. This arrangement consists of a fan installation with intake direct from a large space or room but discharging into a duct. This is the usual blower arrangement and is applicable whether the blower is of the centrifugal or the propeller type.

Arrangement C—Booster. This arrangement provides an intake through a length of duct and a discharge from the fan into a discharge duct. This arrangement is applicable to propeller as well as centrifugal-type fans thus installed.

Arrangement D—Exhauster. In this arrangement the intake is through an inlet duct but the air is discharged directly into a large space. It is applicable to all types of fans thus installed.

Head Definitions

Velocity Head h_v. The definitions of heads may be more readily understood if it is recognized that all heads represent essentially energy per pound of fluid. The velocity head at any point in the system, if expressed in feet of the fluid flowing, is the kinetic energy per pound of fluid at that point. This head expressed in inches of water is proportional to the kinetic energy by a factor involving the ratio of the density of water to that of the fluid flowing. In this investigation the velocity heads at the specified points in the inlet and discharge ducts were calculated from the quantity flowing, as measured by a nozzle installed

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[2] Formerly Balwin Research Fellow, Lehigh University. Jun. A.S.M.E. Mr. Somers was graduated from Lehigh University with a B.S. degree in mechanical engineering in 1933, and received his master's degree in 1935. He was selected for fellowship on graduation and has continued in that position for the past two years. At present he is employed as a cadet engineer with the Babcock and Wilcox Company, Barberton, Ohio.

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

[3] "Influence of Inlet Boxes on the Performance of Induced-Draft Fans," by L. S. Marks and E. A. Winzenburger, Trans. A.S.M.E., vol. 54, 1932, paper FSP-54-16, pp. 213–220. "Influence of Bends in Inlet Ducts on the Performance of Induced-Draft Fans," by L. S. Marks, J. Lomax, and R. Ashton, Trans. A.S.M.E., vol. 55, 1933, paper FSP-55-9, pp. 133–143. "Influence of Bends or Obstructions at the Fan Discharge Outlet on the Performance of Centrifugal Fans," by L. S. Marks, J. H. Raub, and H. R. Pratt, Trans. A.S.M.E., vol. 56, 1934, paper FSP-56-12, pp. 767–771.

TABLE 1 QUANTITY-HEAD CHARACTERISTICS FOR THE INSTALLATION ARRANGEMENTS IN FIG. 1

Installation arrangement	Fan total head, H_T		Fan static head 1st def., H_S		Fan static head 2nd def., H'_S
A—Bare fan	H_{TA}	=	H_{SA}	=	H'_{SA}
B—Blower	H_{TB}		H_{SB}	=	H'_{SB}
C—Booster	H_{TC}		H_{SC}		H'_{SC}
D—Exhauster	H_{TD}	=	H_{SD}		H'_{SD}

NOTE: Characteristics H_{TA}, H_{SA}, and H'_{SA} can be represented by a single curve. Characteristics H_{SB} and H'_{SB} can be portrayed by a single curve. Characteristics H_{TD} and H_{SD} can be portrayed by a single curve. A separate curve is required for each of the following characteristics: H_{TB}, H_{TC}, H_{SC}, H'_{SC}, and H'_{SD}.

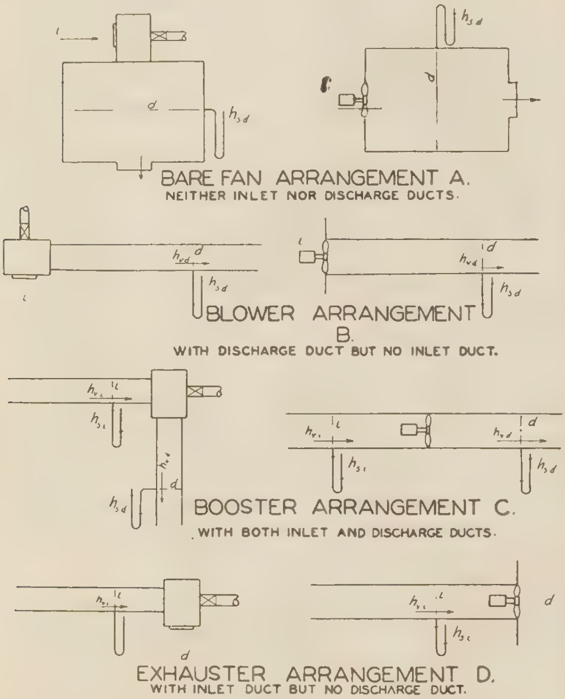

FIG. 1 BASIC ARRANGEMENTS OF FAN INSTALLATIONS

elsewhere in the system, and a knowledge of the duct area at the points in question.

Static Head h_s. Static head, if expressed in feet of the fluid flowing, is the work required per pound to cause the fluid to move against the pressure measured, in fan work, above atmosphere. This static head is equal to the product of the pressure above atmosphere and the specific volume of the air. It has been variously named pressure energy, flow work, displacement energy, and pressure head. This head, if expressed in inches of water, is proportional to the pressure energy by a factor involving the ratio of the density of water to that of the fluid flowing. In this work the static head was determined by observation of the static-pressure gages attached at the specified points.

Total Head h_t. The total head at a point in the system is the sum of the pressure energy and the kinetic energy, both measured at the point in question. If this head is expressed in inches of water, it will be proportional to this sum of energies.

In fan performance we are concerned with certain changes in energy that the fan produces for a given discharge. These increases in energy are the differences between the corresponding heads at the discharge and the inlet. These differences in head will be called the "fan heads," represented by the general symbol H, as distinguished from h, which is the symbol for the head at a certain point.

Of the several possible fan heads which may be obtained from various differences between the inlet and discharge heads, only three have been considered of value in fan work. These three are defined in the following paragraphs.

Fan Total Head. Fan total head is the total head at some specified plane on the discharge side of the fan minus the total head at some specified plane on the inlet side. It can be expressed as

$$H_T = h_{td} - h_{ti}$$

where H_T is the fan total head, h_{td} is the discharge total head, and h_{ti} is the inlet total head.

Fan Static Head—First Definition. The first definition of fan static head stipulates that this head is equal to the discharge static head minus the inlet total head, or

$$H_S = h_{sd} - h_{ti}$$

where H_S is the fan static head as given in this first definition, h_{sd} is the discharge static head, and h_{ti} is the inlet total head.

Fan Static Head—Second Definition. The second definition of fan static head stipulates that this head is equal to the discharge static head minus the inlet static head, or

$$H'_S = h_{sd} - h_{si}$$

where H'_S is the fan static head as given in this second definition, h_{sd} is the discharge static head, and h_{si} is the inlet static head.

With three definitions for fan head and four necessary arrangements to cover various applications, it appears that there are twelve possible quantity-head characteristic curves to completely cover the performance of a fan which has mounting flanges for inlet and discharge ducts, thus permitting it to be installed in any one of the four basic arrangements. However, twelve tests and as many corresponding curves are not required to determine and portray these characteristics.

THE FAN TESTS

In order to determine how the characteristics vary for the different arrangements, a fan was set up and tests conducted for all four basic arrangements, and the twelve characteristics computed. These characteristics are analyzed, both to evaluate their practical utility, and to determine the minimum number and character of the tests required to establish those characteristics which are considered to be of value. This leads to a definition of the quantity-head characteristic.

The tests were run on a single-inlet Clarage exhauster having forward-tilting blades. The inlet and discharge openings were both 9 in. in diameter. The fan was driven through a flexible coupling by a squirrel-cage induction motor of 1800 rpm synchronous speed.

The inlet and discharge ducts used were 9 in. in diameter and made of sheet metal, the joints being riveted and soldered to make them airtight. The inlet duct was 6 ft, or 8 diameters, long and the discharge duct was 10 ft, or about 13.5 diameters long. Brass bosses, drilled and tapped for $^1/_8$-in. pipe, were soldered on each duct for the static-pressure taps at points three-quarters of the duct length from the duct inlet. All burrs were removed from the edges of the static tap holes in the ducts.

All flow measurements were made with a 260-mm 1912 V.D.I. nozzle, using a coefficient of 0.97, placed at the outlet of a large test chamber 28 ft long and 33 in. in diameter. When the discharge duct was connected to the fan, the air was discharged from the duct into this test chamber, the control of the flow being made by means of a series of throttle plates inserted at the connection between the duct and the chamber as shown in arrangement B of Fig. 2. When no discharge duct was used the fan discharged directly into this test chamber as shown in ar-

rangement *A* of Fig. 2. Flow control was then effected by means of a multihole throttle plate inserted 10 ft downstream in this chamber. This plate also acted as a screen to straighten the flow for the nozzle approach.

In order to produce the four basic installation arrangements with the equipment described, it was set up as follows in the four test arrangements shown in Fig. 2.

Test Arrangement A—Bare Fan. The fan intake was directly from the room, and the discharge was directly into the test chamber.

Control of flow was effected by means of the multihole throttle plate 10 ft downstream in the test chamber.

Test Arrangement B—Blower. Intake was directly from the room, and the discharge was made through a discharge duct from which the air passed into the test chamber. The multihole throttle was removed, the chamber acting only as a device for smoothing the approach to the nozzle. Flow was controlled by means of the series of throttle plates at the point where the duct joins the test chamber.

Test Arrangement C—Booster. The intake was from the room through an inlet duct, and the discharge was through a discharge duct from which the air passed into the test chamber. Flow was again controlled by the series of throttle plates at the end of the discharge duct.

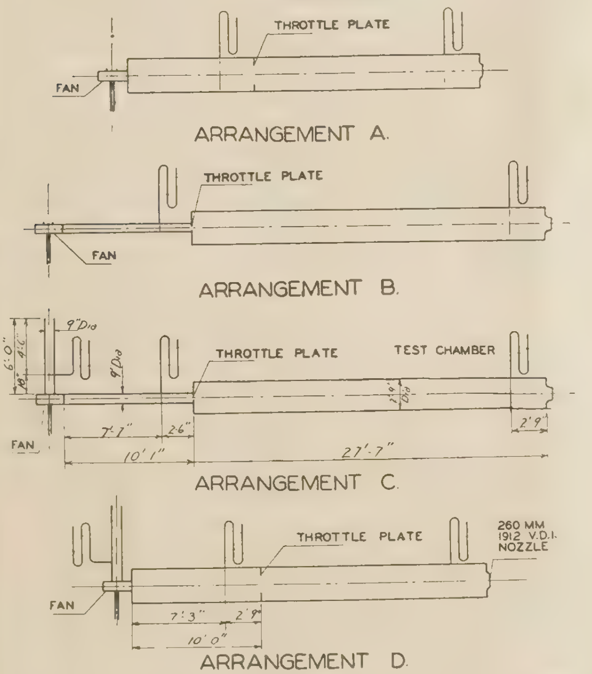

Fig. 2 Fan Test Installations of Basic Fan Arrangements

Test Arrangement D—Exhauster. The intake was again from the room through an inlet duct and the discharge was directly into the test chamber. The multihole throttle plate was used for flow control.

The inlet static head was measured, when required, by means of an inclined draft gage, read to 0.005 in. of water. The discharge static head was measured by a vertical U-tube water manometer. The velocity heads at the points at which the static heads were measured were calculated from the flow measured by the nozzle and the areas of the ducts at these points. When the fan discharged directly into the test chamber, the discharge pressure was read at a point 33 in. upstream from the throttle plate, or 2.7 chamber diameters after the discharge into the chamber. The nozzle differential was measured by means of an inclined-tube draft gage read to 0.001 in. of water. All draft gages were calibrated by means of a hook-gage manometer. Corrections were made to standard density, 0.07488 lb per cu ft, and to the fan speed of 1800 rpm. No measurements were made of power, since the scope of the investigation was limited to the study of the quantity-head characteristics.

Tests were conducted, giving six points, evenly distributed, on each characteristic curve for each of the four basic arrangements.

Analysis of Data

Fig. 3 shows the 12 quantity-head characteristics obtained from this fan by applying the three definitions of fan head to the four

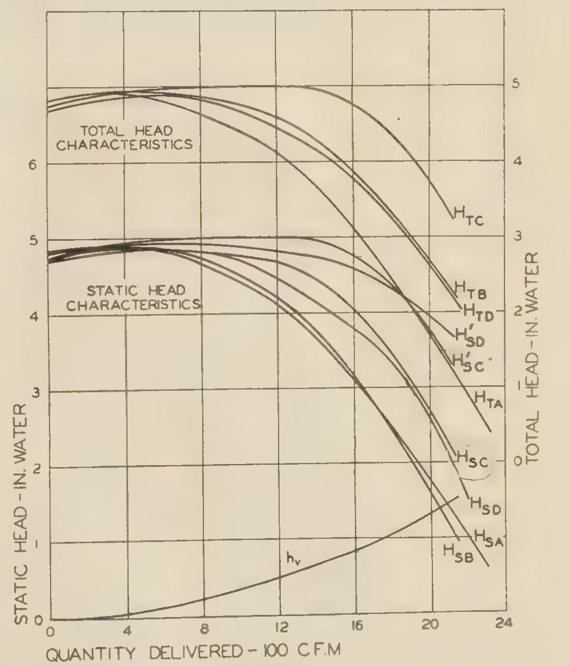

Fig. 3 Quantity-Head Characteristics of an Exhauster Fan

basic arrangements. The upper four curves are the total-head characteristics, while the lower group of curves are the static-head characteristics. These curves show the striking differences which exist between the characteristics for the different arrangements and for the various definitions of fan head. These differences show vividly the necessity for a very careful analysis of the definition of the quantity-head characteristic of fans. This is especially appreciated when it is recognized that these diverse curves represent results which may be obtained for a single fan by use only of definitions and arrangements which occur commonly in practice.

Before presenting a detailed analysis of the characteristics for each basic arrangement, a general principle will be stated relative to the places at which heads are measured or expressed. In any arrangement or definition, heads existing at the inlet must be measured at some specified plane on the inlet side, and heads at the discharge must be measured at some specified plane on the discharge side. On the discharge side of a blower for example, a procedure of adding heads measured at two different places on the discharge side is utterly untenable. The discharge total

head must be obtained by adding velocity and static heads, *both measured at the same plane.* This principle has been violated unfortunately, on occasion, in practice.

In arrangements where ducts are applied at the fan inlet or discharge, the heads are to be measured in the duct at a plane so located that the flow conditions have some measure of stability and uniformity. Where no duct is applied, the plane for head measurement may not be directly at the entrance to or exit from the rotor, because the uncertain flow conditions render impossible either the measurement or computation of heads at these places. The heads must be measured in these cases, at planes sufficiently removed from the fan inlet or discharge such that the kinetic energy at these planes is sensibly zero. In these circumstances the total head equals the static head.

Having established the places at which heads must be measured, indicated at i and d in Fig. 1, we may now proceed with the analysis of the characteristics as obtained for each basic arrangement.

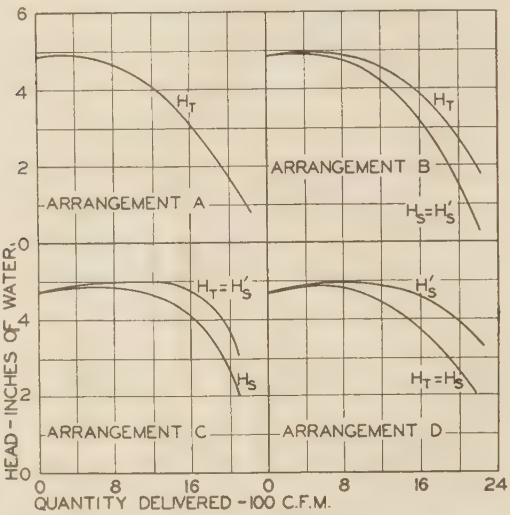

Fig. 4 Quantity-Head Characteristics of an Exhauster Fan Shown by Basic Fan Arrangements

Arrangement A. It is recalled that arrangement A is for the bare fan, of either propeller or housed type, with no inlet or discharge ducts attached. The inlet total head, measured at i, Fig. 1, is equal to the inlet static head since the velocity head is sensibly zero at the plane of head measurement. The discharge total head, measured at d, is likewise equal to the discharge static head. Therefore, the single characteristic curve, shown in Fig. 4, represents the quantity-head characteristics of arrangement A for all three definitions of fan head.

Arrangement B. In arrangement B, which has a discharge duct but no inlet duct, the inlet heads are determined in open space at a plane i, Fig. 1, and the discharge heads are measured in the discharge duct at a plane d. The inlet static head will be equal to the total head. The discharge total head will be greater than the discharge static head by the amount of the discharge velocity head, all measured at plane d. This difference will cause the total-head characteristic to be higher than the static-head characteristic by the discharge velocity head, as shown in Fig. 4. However, the two definitions of fan static head give the same characteristic curve.

Arrangement C. As shown in Fig. 1, both inlet and discharge ducts are used in arrangement C. Therefore, the inlet heads are to be determined in the inlet duct at a plane i, and the discharge heads are to be determined in the discharge duct at a plane d. The inlet total head is greater than the inlet static head by the amount of the inlet velocity head. The discharge total head is greater than the discharge static head by the discharge velocity head. These circumstances cause the static-head (first definition) characteristic to be lower than the total-head characteristic by the discharge velocity head, as shown in Fig. 4. The second definition of fan static head is equivalent to adding the inlet velocity head to the fan head obtained by the first definition. This would, in general, make necessary three distinct characteristic curves applying to this arrangement. For this particular fan, however, the inlet velocity is the same as the discharge velocity, since the two ducts are of the same area. This makes the characteristic curve obtained by the second definition of fan static head coincide with the characteristic obtained by use of the fan total head.

Arrangement D. In arrangement D, it is recalled that there is an inlet duct but no discharge duct. The inlet heads are determined in the inlet duct at a plane i, while the discharge heads are measured at plane d, in open space at a distance sufficiently removed from the fan outlet that the velocity head is sensibly zero. The inlet total head and inlet static head, measured at plane i, differ by the inlet velocity head. The discharge total head is identical with the discharge static head, both measured at plane d, since the discharge velocity head is zero. Thus, the total-head characteristic and the static-head characteristic given by the first definition of fan static head will coincide, while the static-head characteristic given by the second definition of fan static head will be greater than these two by the inlet velocity head.

The various quantity-head characteristics for the four basic installation arrangements present the differences and equalities summarized in Table 1. It is seen from this table that a total of eight curves is required for the three head definitions and for the performance of fans arranged in the four basic installations shown in Fig. 1.

Characteristics of Various Fan Types

The foregoing analysis applies to fans provided with mounting flanges for inlet and discharge ducts, and thus adaptable to four possible installation arrangements. All fans do not have flanges to receive inlet and discharge ducts, however, and are thereby subject to certain limitations. The possible installation arrangements, the required test arrangements and the resulting characteristic curves for various fan types may now be established.

All fans may be classified under the four following types:

1 Fans with no mounting flanges for inlet or discharge ducts. Wall-type propeller or disk fans are frequently of this type.

2 Fans with no inlet-duct mounting flanges, but with discharge-duct mounting flanges. This is the common blower-type fan.

3 Fans with mounting flanges for both the inlet and discharge ducts.

4 Fans with inlet-duct mounting flanges, but no discharge-duct mounting flanges.

A wall-type propeller fan which is provided with no connections for ducts must be classified under type-1. It must be tested under arrangement A, and since this arrangement provides the only possible method for mounting such a fan, the single characteristic curve fully defines its performance.

A usual-type blower, having flanges for mounting only a discharge duct, is an example of type-2. The performance of this type fan for all its possible installation arrangements can be defined fully by three characteristic curves determined under test arrangements A and B.

Table 2 presents for each type of fan (a) all possible installa-

TABLE 2 NUMBER OF ARRANGEMENTS, CHARACTERISTICS, AND REQUIRED CURVES FOR VARIOUS FAN TYPES

Fan type	Possible installation arrangements	Necessary test arrangements	Number of characteristics	Number of curves Per instal.	Per type
1 No ducts........	A	A	3	1	1
2 With discharge-duct flange only.......	A	A	3	1	3
	B	B	3	2	
3 With flanges for discharge and inlet ducts............	A	A	3	1	8
	B	B	3	2	
	C	C	3	3	
	D	D	3	2	
4 With inlet-duct flange only.......	A	A	3	1	3
	D	D	3	2	

tion arrangements, (*b*) the corresponding necessary test arrangements, (*c*) the number of characteristics possible for each installation arrangement, (*d*) the number of curves required to portray the characteristics for each installation arrangement, and (*e*) the total number of curves required to portray all the characteristics possible for a given type of fan. The most significant deduction to be made from Table 2 is that from one to eight curves are required to portray the possible characteristics for the various fan types. It must be remembered that these characteristics are in strict accordance with the three currently used definitions of fan head as applied to the installation arrangements for which each type is adaptable.

Development of Fan-Head Definition

The purpose of a fan characteristic is to delineate useful and direct information regarding how the fan will perform in its installation. Specifically, the quantity-head characteristic shows the quantity of air which will be delivered against a certain resistance in the intended installation. It now becomes necessary to evaluate the three fan-head definitions to discover which of these truly contribute useful information toward this objective.

The fan-head definitions are each made up of various combinations of static and total heads at inlet and discharge. Let us first dispose of the heads on the inlet side. It will be shown that the only head on the inlet side which contributes a useful function in defining the fan characteristic is the total head.

Referring to the arrangement with no inlet duct, Fig. 5, let the inlet total head be measured at some point P at a distance from the fan inlet, the velocity being negligible at that point. Then let the static and velocity heads be measured at a point Q very close to the fan inlet, no inlet duct being used. At point P all of the energy is in the form of static-head or pressure energy. There are practically no losses in energy in the air in flowing from P to Q. The gain in kinetic energy at Q is at the expense of the pressure energy at Q. Expressed as a simple relation

$$h_{tP} = h_{sP} = h_{vQ} + h_{sQ}$$

Now let an ideal inlet duct (no inlet or wall-friction losses) be attached to the fan inlet. Let P be located outside the duct, again at a point of negligible velocity, and let Q be located inside the duct. Under these circumstances we again find that the total head remains constant from P to Q regardless of the kinetic-energy increase at Q. Thus, the fan, either without any inlet duct or with an ideal inlet duct, does not add any energy to the air before the air actually enters the fan rotor. This is counter to some current conceptions. For example, a fan draws air from the atmosphere. Then at point P, the total head and static head are both zero. At point Q the sum of the static and velocity heads are also zero. Any kinetic energy which the air possesses at fan inlet was not produced by the fan, but at the expense of a corresponding decrease in pressure energy in the air stream. The fact is, that any energies existing at the fan inlet are supplied *to* the fan and not *by* the fan.

The ideal inlet duct is now replaced by an actual inlet duct. Because of the inlet and wall-friction losses the total head at Q will now be less than at P. The energy which goes to the fan at the fan inlet is still exactly and only the total energy in the air at the inlet, regardless of losses or other conditions which exist up to the fan inlet. The conclusion is that in any definition of fan head, we must debit to the fan the energy supplied to the fan by the air at the fan inlet. This energy supply is the inlet total head and is equivalent to h_{si} for fans without inlet ducts, and to $(h_{si} + h_{vi})$ for fans with inlet ducts.

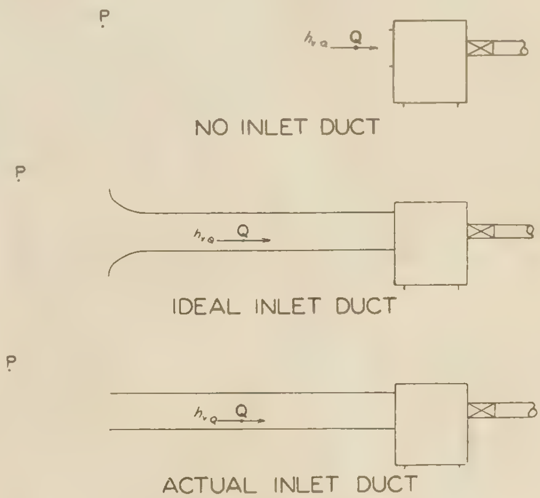

Fig. 5 Heads at Inlet to Fan

This eliminates the second definition of fan static head used throughout this paper, which erroneously credits the fan with the inlet velocity head.

Let us now consider the heads at the discharge side of the fan. In general, the resistance of the external system into which the fan discharges air is overcome by the energy of the discharge pressure. The only way by which the discharge kinetic energy of the air may have utility in overcoming resistance is by the introduction of a diffuser which converts kinetic energy into pressure energy. If the diffuser is applied at the fan discharge it should be considered as part of the fan and the discharge static head which is measured in this case would include the recovered kinetic energy at the fan outlet. If, on the other hand, the diffuser is installed at some point in the external system, it becomes a part of that system and any conversion of energy made by use of the diffuser cannot be properly credited to the fan. The authors, therefore, propose that in defining the fan head the discharge velocity head be entirely disregarded and the discharge static head only be used.

Consider two fans X and Y, having equal discharge total energy for equal quantity discharged, but fan X having a much larger portion of kinetic energy. These fans are not of equal utility for purposes of overcoming resistance. The purposes of factors such as good housing design and large outlet area are to convert kinetic energy to pressure energy. In most cases, if not in all cases, the fan should not be credited with unconverted kinetic energy.

We are now in a position to develop a definition of fan head. The fan head which gives the most useful information regarding fan performance in its installation is one which measures the

energy increase from the energy level of the air at the fan inlet to that energy level at the fan discharge which is directly usable, that is, without further conversions, in an external system. We have shown that at the fan inlet, the fan must be debited with both the pressure and kinetic energy existing there, while at the discharge of the fan, only the pressure energy may be properly credited to the fan.

The foregoing requirements for a fan-head definition are met precisely by the first definition of fan static head used throughout this paper. Specifically, *the fan head to be used in the quantity-head characteristic is defined as the discharge static head minus the inlet total head.* In arrangements which have discharge ducts, the discharge static head is measured in the duct at a specified plane so located that the flow conditions have some measure of uniformity. In arrangements without discharge ducts, the discharge static head is measured in the space into which the fan discharges. In arrangements which have inlet ducts, the inlet static and velocity heads are measured in the inlet duct at a specified plane so located that the flow conditions have some measure of uniformity. In arrangements without inlet ducts the inlet total head is the static head measured in the space from which air is supplied to the fan.

TABLE 3 NUMBER OF ARRANGEMENTS, CHARACTERISTICS AND CURVES FOR VARIOUS FAN TYPES, BASED ON STANDARD DEFINITION OF FAN HEAD

Fan type	Possible installation arrangements	Necessary test arrangements	Number of characteristics	Number of curves Per instal.	Number of curves Per type
1 No ducts........	*A*	*A*	1	1	1
2 With discharge-duct flange only.......	*A*	*A*	1	1	2
	B	*B*	1	1	
3 With flanges for discharge and inlet ducts.........	*A*	*A*	1	1	4
	B	*B*	1	1	
	C	*C*	1	1	
	D	*D*	1	1	
4 With inlet-duct flange only.......	*A*	*A*	1	1	2
	D	*D*	1	1	

Using this definition of fan head, the complete schedule of fan types is given in Table 3, with all possible installation arrangements and corresponding necessary test arrangements, and the number of characteristics and curves required to completely portray the performance of each fan type.

From Table 3, it may be seen that for type-1, only one characteristic is possible, while for type-3, four characteristics and four curves are necessary to cover the four possible installation arrangements. The fan classified as type-3 is reported upon in this paper, the four characteristic curves being the four lower static-head characteristics shown in Fig. 3. Incidentally, the discharge velocity head, which is not included in the fan head, may be conveniently and usefully portrayed by plotting h_v vs Q, as shown in Fig. 3. This curve is identical for all possible arrangements which have discharge ducts, depending as it does only on the quantity discharged and the area of the discharge duct.

Quantity Measurement

A word need be said as to the measurement of the quantity term in the quantity-head characteristic. First, it should be recognized that by no means must the quantity be measured at the point at which a head is measured. The quantity may be measured at any convenient place in the system. Further, depending upon the test arrangement indicated for the particular cases, various acceptable methods of quantity measurement become more convenient and economical. The pitot tube, nozzle, and orifice are among the acceptable methods of measurement of air quantities in fan testing. Conditions of accuracy of these methods are under continual study by experimenters, and are eventually set forth in codes. In arrangements without inlet or discharge ducts, the use of nozzles or orifices in the walls of a large test chamber becomes an accurate, convenient, and economical method. In arrangements using small inlet or discharge ducts, nozzles or orifices at the entrance to inlet ducts or at the exit from discharge ducts are satisfactory. In very large ducts, nozzles, and orifices become more expensive, and the pitot tube is perhaps more satisfactory. In moderate-size ducts, the choice from among the acceptable methods of air measurement may be based upon considerations of convenience and availability of materials.

Definition of Quantity-Head Characteristic

The definition of fan head to be used in determining the quantity-head characteristic of a fan has been established. The quantity term is not subject to definition, but must be measured by some acceptable method. The quantity-head characteristic may now be defined as the relation between the quantity discharged and the defined fan head for a specified installation arrangement. Fan head is defined as the discharge static head minus the inlet total head.

It has been pointed out that for certain type fans there are possible more than one characteristic as defined by the established fan-head and quantity measurements, depending on the number of possible installation arrangements to which the fan may be adapted. Each particular fan type, however, is designed primarily for use in a particular installation arrangement which may be termed its *primary arrangement.* That characteristic which is obtained by use of the test arrangement corresponding to the primary installation arrangement of each type fan may be termed the *primary characteristic* for each type. For general use, then, this primary characteristic only need be determined, but with the understanding that this characteristic will not necessarily apply when the fan is installed in any arrangement other than its primary one. Thus, for fan type-1, with no mounting flanges for inlet or discharge ducts, and thus having only one possible arrangement, the only possible characteristic is the primary one. For the fan classified as type-2, the ordinary blower with only a discharge-duct mounting flange, and thus having two possible arrangements, the primary arrangement is considered to be that with a discharge duct attached. The primary characteristic will be that obtained using test arrangement *B*, the blower arrangement with the discharge duct attached. The primary arrangement of a fan classified as type-3 having mounting flanges for the inlet and discharge ducts, is that in which both inlet and discharge ducts are used. The primary characteristic is, therefore, that obtained by use of test arrangement *C*, which has both ducts attached. The primary arrangement for type-4 is that in which the inlet duct is attached. The primary characteristic is that obtained by test arrangement *D*. Thus, for any type of fan there may be established a primary arrangement and a single corresponding primary characteristic.

For the fan reported on in this paper, using the several fan-head definitions in common use and the four possible arrangements, 12 characteristics were originally determined and shown. By use of the established definition of fan head, the possible characteristics reduce to four. Of these four characteristics that one is considered the primary characteristic which shows the performance of the fan when installed in its primary arrangement which, for this fan, is the one with both inlet and discharge ducts in place. This characteristic is shown by the curve marked $H_{S,C}$ in Fig. 3. If the fan is to be installed in any other of the four possible arrangements it may not be taken for granted that this primary characteristic applies, but a separate test must be conducted to determine the characteristic for the intended arrangement.

Drying Problems of the Ceramic Industry

By JOHN L. CARRUTHERS,[1] COLUMBUS, OHIO

The purpose of this paper is to outline and discuss the various problems encountered in designing and operating ceramic-drying systems, and to indicate the several methods that may be used in attacking the more difficult problems that arise.

The process of drying clay wares may be a relatively simple one or it may be quite complex. This is due to (1) wide variations in production requirements in different industries, (2) extreme differences in the physical characteristics of clay bodies used, and (3) various sizes and shapes of different products. In many cases, the methods and equipment used may appear to be very slow and inefficient when considered in the light of modern drying practice. Their use is generally justified, however, when all factors are taken into consideration. However, this does not imply that all present systems are best suited for their particular task.

The problems discussed in the paper center around the development, maintenance, and determination of safe, rapid, and economical drying rates and the reduction or prevention of ware losses in order to increase drying and manufacturing efficiencies.

IN MANUFACTURING some wares, drying is immediately followed by a firing process, which in its early stages could function as a drying process, since there is a supply of heat and a suitable means for removing water vapor. Vitreous floor tile and small porcelain wares are commonly dried in the kiln, because the pieces are small and fragile and must not be subjected to unnecessary handling. However, a drying process is ordinarily required for the following reasons: (1) To give ware sufficient strength to be handled in loading kilns. (2) Kiln drying increases the time of the firing cycle thus requiring additional equipment. (3) There is danger of the ware being marred by kiln scum which forms if sulphur gases come in contact with water in the surface layers of the clay ware. (4) The ware must be dried before the glaze can be applied. (5) The temperature and humidity of the kiln gases cannot be readily controlled within the desired limits for satisfactory drying conditions.

Requirements of the Drying Process

The most important requirement is safe drying which means that the ware from the drier will be free of any defects that may be caused by improper drying conditions. Any ware spoiled during drying is practically worthless except in the few cases where cracked ware can be patched or repaired. Expenditures on spoiled ware are lost and the salvaged clay is of little value.

Clay wares should ordinarily be dried rapidly and uniformly and to the hygroscopic or regain water content (2 to 4 per cent). However, in some cases complete drying is not desired, because the ware may be too brittle for handling when bone dry, or it is too expensive to remove the last small percentage of water from the ware while in the drier. Also, several partial drying processes are used in the industry, e.g., drying of filtered clay for dry pressing and conditioning of clay blanks for turning, repressing, or "sticking."

Ware Forming Methods

A consideration of the methods used to form ceramic wares is necessary in the solution of drying problems, since some of the physical characteristics of the plastic and dry pieces will be affected by the method used. Following are brief descriptions of forming methods with lists of products indicating some of the conditions encountered:

1 *Dry Pressing.* A slightly moistened or damp clay body in dust form (water content 6 to 15 per cent) is placed in a steel mold and then compacted by the application of pressure to the die. The pressure may vary from several hundred to several tons per square inch. Products: Floor and wall tile, low-tension electrical porcelain, special refractories, and fire-clay bricks and shapes.

2 *Stiff-Mud Extrusion.* A semiplastic clay body (water content 15 to 25 per cent) is forced through an orifice or die which controls the dimensions of the cross-section of the piece. The extruded column is cut into required lengths by steel wires. Products: Common, face, glazed, paving, and fireclay brick; hollow building tile; telephone conduit; sewer pipe; drain tile; roofing tile; spark-plug cores; machine-made terra cotta; and blanks for electrical porcelain bushings.

3 *Soft-Mud Forming.* A soft plastic clay body (water content 20 to 30 per cent) is either modeled by hand or is pressed by hand or by mechanically operated mandrels into plaster, wood, or steel molds. Wares made in plaster molds are partially dried in the mold so that they may be easily removed and also to develop sufficient strength for handling and finishing. Products classified by name of process are: (*a*) Hand modeling—terra-cotta statuary, saggers, glass pots, and pottery. (*b*) Throwing on potter's wheel—art pottery and specialties. (*c*) Jiggering (plaster molds)—dinnerware, stoneware, chemical porcelain, and art ware. (*d*) Hot pressing (plaster molds)—electrical porcelain. (*e*) Machine pressing (plaster and steel molds)—roofing tile, flower pots, saggers, and crucibles. (*f*) Hand pressing (plaster molds)—terra cotta, faience tile, and chemical stoneware. (*g*) Hand and machine pressing (wood molds)—common and fireclay brick, and refractory shapes.

4 *Casting.* A clay-water mixture of the consistency of cream (water content 20 to 40 per cent) is poured into plaster molds. The clay is deposited on the side walls of the molds as the water is absorbed by the plaster. The use of electrolytes in the mixture makes it possible to use relatively small amounts of water. The ware must dry to a rigid condition by absorption while in the mold to permit removal. Products: Hollow shapes in dinnerware, art pottery, saggers, and sanitary ware.

5 *Additional Forming Methods.* Other methods may be required

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to complete the piece only partially formed by one of the preceding operations. These methods either change the structure of the clay body or develop other conditions which may influence the drying operations. The methods are: (*a*) Repressing—applied to stiff- and soft-mud products, such as paving and fireclay brick. Preliminary drying is necessary with soft-mud products. (*b*) Finishing or tooling of surfaces—applied principally to wares formed in molds. (*c*) Turning on a lathe—spark-plug porcelains and electrical insulator bushings. (*d*) Mitering or "Sticking"—two or more pieces formed separately are combined by pressing the adjoining surfaces together, using a very soft plastic clay or clay slip as the bonding agent. This method is used for cup handles, sewer-pipe fittings, closet bowls, and terra-cotta corner pieces.

Drying Problems

The various problems of ceramic drying comprise: (1) Elimination of ware losses in drying; (2) developing faster safe rates of drying for clay bodies and shapes; and (3) developing methods for determining, experimentally, the safe drying time.

Elimination of Ware Losses. The seriousness of this problem depends upon local conditions. Ware losses may be due to cracking, checking of surfaces, warping, strains, slumping, efflorescence or "drier scum," blistering, or other troubles. The causes of these troubles may be in the drying system or may be attributed to raw materials or processing. There may be a lack of control in the system preventing the maintenance of uniform drying conditions, or there may be changes in the properties of the raw materials. Also, processing conditions are encountered that markedly influence the drying behavior of the ware. Unless such changes can be anticipated, it is impossible to know that drying conditions should be altered until after a considerable amount of ware has been spoiled.

Cracking, checking, warping, and strains in the ware are all due to drying at excessive rates, which set up too extreme variations in shrinkage throughout the piece and prevent an adjustment of the stresses. The actual fault developed will depend upon the degree of localized shrinkage, the shape and position of the piece, and the strength of the clay body. The factors influencing the rate of drying and the development of these troubles are humidity of the air; velocity of air; direction of air flow; temperature of the air and ware; size and shape of the piece; diffusion and shrinkage characteristics of the clay body; water content; and strength of the clay body.

Slumping is due to the absorption of water condensed from humid air on the surface of the piece. Also, plastic clay wares may slump if they are heated in a humid atmosphere that permits little, if any, drying. The increase in temperature lowers the viscosity of the water, and the clay-water mixture cannot support its own weight. Blisters are caused by air entrapped in surface layers of the piece. If the clay is softened by condensation and the air expanded by heating, then a blister will be formed. Efflorescence or "drier scum" is due to either soluble salts in the clay, brought to the surface during drying, or formed by the reaction between sulphur gases in the drier chamber, water and minerals in the clay. Other items that may cause a loss of ware are vibrations of conveying equipment, poor handling of the ware, or lack of provisions for shrinkage when setting the piece.

There are three general methods which may be used to solve the problem of ware losses. The choice of method will depend upon the results of an analysis of the prevailing conditions and the weight of the various factors. The mode of attack may be along any of the following lines:

(*a*) The use of a drier that may be closely controlled over a fairly wide range of drying conditions and changes in atmospheric conditions. This may require remodeling or replacing existing driers.

(*b*) The use of a control system which will eliminate changes in the drying system or the drying behavior of the clay ware. Ware of uniform characteristics should be delivered to the drier. However, possible variations in clays and processing conditions beyond the control of the drier or its operator often prevent satisfactory drying.

(*c*) Change the physical properties of the clay body, where possible, so that the ware can be safely dried under conditions that can be economically maintained in the existing equipment. This method is discussed in more detail in the next section.

2 *Development of Faster Safe Drying Rates.* A reduction in the excessive safe drying time required in many clay plants would have several obvious advantages. The solution of this problem is by improving drier designs and techniques or the application of ceramic technology. Research work in ceramics during the past ten years has made available new information on the nature of clays and methods of varying them, especially in respect to their drying behavior. Consequently any of these methods, or others, may be the most desirable one to use, providing the drier equipment is satisfactory.

Rapid drying is not always desirable from an economic standpoint, especially with wares such as dinnerware, stoneware, or art pottery. In some plants, a supply of partially fabricated ware is maintained at several stages of the manufacturing process in order to meet fluctuations in production orders. Consequently, storage rooms in which the ware is dried slowly are more economical than regular driers since this procedure eliminates an extra handling. Also, there are some types of ware, such as glass pots, which seem to have better service qualities if they are dried slowly.

Changes in drier design or operation quite often make it possible to dry ware safely at rates that would be disastrous with the existing system. Cut-and-try methods may be required to obtain the desired results due to conditions often difficult to explain. Several features of design or methods of operation, that might be investigated in the remodeling of old driers or the design of new ones, are as follows:

(*a*) Direction and velocity of air flow. This feature is quite vital with many products. Whenever possible, all surfaces should have uniform treatment.

(*b*) Location of heating units. Rapid safe drying has been obtained in some driers by placing the steam-radiator elements in the drying chamber. This arrangement eliminated the necessity for a large recirculating-air volume to deliver the required units to the ware.

(*c*) Heating the ware before placing it in the drier or heating it (with little drying) in a controlled humid atmosphere at the start of drying, will ordinarily make possible the use of increased drying rates.

(*d*) Provide a uniform air circulation throughout the ware setting, especially in continuous driers. If all ware does not receive the proper initial treatment due to poor circulation, then the rapid drying conditions later on will damage that ware not properly treated. In some cases the ware setting in the drier may require adjustment.

(*e*) Adjust air volumes and temperatures to prevent condensation of water vapor on the ware.

(*f*) Both continuous and intermittent drying systems have definite applications in the industry. A change from one type to another is sometimes indicated.

There are several methods, based upon application of ceramic technology that may be used to increase the safe drying rate of a clay or make it possible to dry certain clays satisfactorily for commercial use. These methods alter various physical properties that govern or influence the rate at which the clay may be safely dried. The properties are shrinkage, pore structure, and

strength which are dependent upon the grain size, the amount and state of colloids of the clay, the water content, and the nature of the minerals in the clay. Briefly, the methods are: (*a*) The addition of a non-plastic material or a less-plastic clay to decrease shrinkage in drying. (*b*) Preheating the clay at temperatures between 400 and 500 F to alter the character of the colloids and thus decrease shrinkage in drying. (*c*) Coarser grinding of materials to decrease shrinkage. (*d*) Weathering of raw clay, additional tempering, aging the plastic body, addition of an organic bond, or the addition of a more plastic clay may be used to increase the strength of the clay. (*e*) A small amount of sodium chloride (NaCl) added to a clay has been effective in several cases. (*f*) The addition of an electrolyte to flocculate the colloidal material. This increases the strength of the body and apparently increases the rate of water diffusion. (*g*) A chemical treatment to cause an exchange of bases in the colloidal material.

In some cases, drier troubles are due to the development of planes of weakness or strains in the pieces by faulty forming processes. The presence of auger, die, or hand laminations, pressure planes, etc., will require the use of slow drying rates to prevent heavy ware losses.

Determination of Safe Drying Rates. At the present time, there is no correlation between the drying behavior of trial pieces under controlled laboratory conditions and that of actual pieces of ware in commercial driers. A solution of this problem is needed for advancing the art of drying clay wares. The determination of required drying rates in commercial driers is a very expensive proposition.

Considerable work has been done both in this country and abroad to obtain more information in regard to the nature of clays and the forces or agencies that govern or influence phenomena such as shrinkage, diffusion, vapor pressures. Other studies have been made to determine what relationships there might be between humidity, temperature, pressure, water content, shrinkage, or clay characteristics and drying behavior. While much valuable information has been obtained, the methods now available for determining the safe drying time of a clay when formed into a definite shape are of a cut-and-try nature. The problem is exceedingly complex since there are many variable factors that must be considered.

Summary

The problems discussed center around the development, maintenance, and determination of safe, rapid and economical drying rates and the reduction or prevention of ware losses in order to increase drier and manufacturing efficiencies.

The principal factors involved in the solution of the problems are: (1) Requirements for drying systems; (2) manufacturing methods employed; (3) the physical characteristics of clays and product shapes; (4) ordinary phenomena of drying such as temperature, humidity, air flow (direction and velocity), heat transfer and vaporization; and (5) economic considerations.

The methods that may be used in solving the existing problems include the following:

(1) Provide correct drying conditions for the ware by remodeling or replacing existing driers so as to eliminate ware losses or increase safe drying rates. Also, provide suitable control methods and apparatus.

(2) Improve the drying behavior of the clay bodies by altering their physical properties so as to prevent ware losses and to increase the safe drying rate. This general method can only be used when the quality of the finished product is not adversely affected. The methods that may be employed are (*a*) the use of non-plastic or less-plastic materials, (*b*) chemical treatment, (*c*) the addition of stronger clays or organic bonds, (*d*) the use of additional processes such as preheating, weathering and aging, and (*e*) varying the methods used in grinding and tempering the clay.

There are plants that are continually encountering many of the difficulties described, while others seldom have drier troubles. However, the industry as a whole encounters these problems continually and only by recognizing that there are problems can improvements be made.

It is believed that the art of drying ceramic wares in the major portion of the industry is on a fairly high level when consideration is given to the complexity and wide variations in the composition and properties of clays, the variety of products made, the different processes used in manufacturing, and the science of drying.

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Discussion

A New Basis for the Rating of Roller-Chain Drives[1]

Norman Arnold.[2] Refinements in methods of design are handicapped in their adoption unless the mathematical work introduced by the change can be made simple. At the suggestion of Professor Bartlett, an effort has been made to devise a simple graphical method for performing the computations on the basis set forth in his paper.

The method here developed employs an alignment chart with a sliding scale, a device that is not as common among engineering computing diagrams as its convenience and simplicity warrant. Attention is called to the three mathematical conditions which must be satisfield. They are obtained from Equations [22] and [27] and Table 1 of the original paper:

$$x = \frac{1000 \text{ hp}}{N_1 N_2} \gtreqless 23\, PB \qquad [22]$$

$$y = \frac{1000 \text{ hp}}{N_1 n_1} \gtrless 12.7\, PBd \qquad [27]$$

$$z = n_1 \gtreqless \text{rpm} \qquad \text{Table 1}$$

The right-hand members of these inequalities involve pitch, bushing length, wt per ft, etc., all of which are constants for any particular standard chain. The author supplied tabulated values of the right-hand members for the various standard chains; and these values were used in preparing the chart.

It will be noted that 1000 hp/N_1 is contained in both Equations [22] and [27], therefore, in preparing the alignment chart, this division can be performed first, and the result used in solving both Equations [22] and [27]. On the accompanying chart, Fig. 1, this division is carried out by drawing a straight line through the points on the scales for N_1 and hp to the Q-axis, (dashed line A in the figure). The inequality 22 is satisfied by any chain whose number appears on the X-scale above the index line from the point on the Q-axis to the value of N_2 (line B in the figure). In general, the conditions limiting the rpm are satisfied by chains whose numbers are nearer the bottom of the chart than the given rpm, n_1; and inequality 27 is satisfied by chains whose numbers are on the Y-scale above the index line from the

[1] Published as paper MSP-57-1, by G. M. Bartlett, in the April, 1935, issue of the A.S.M.E. Transactions.

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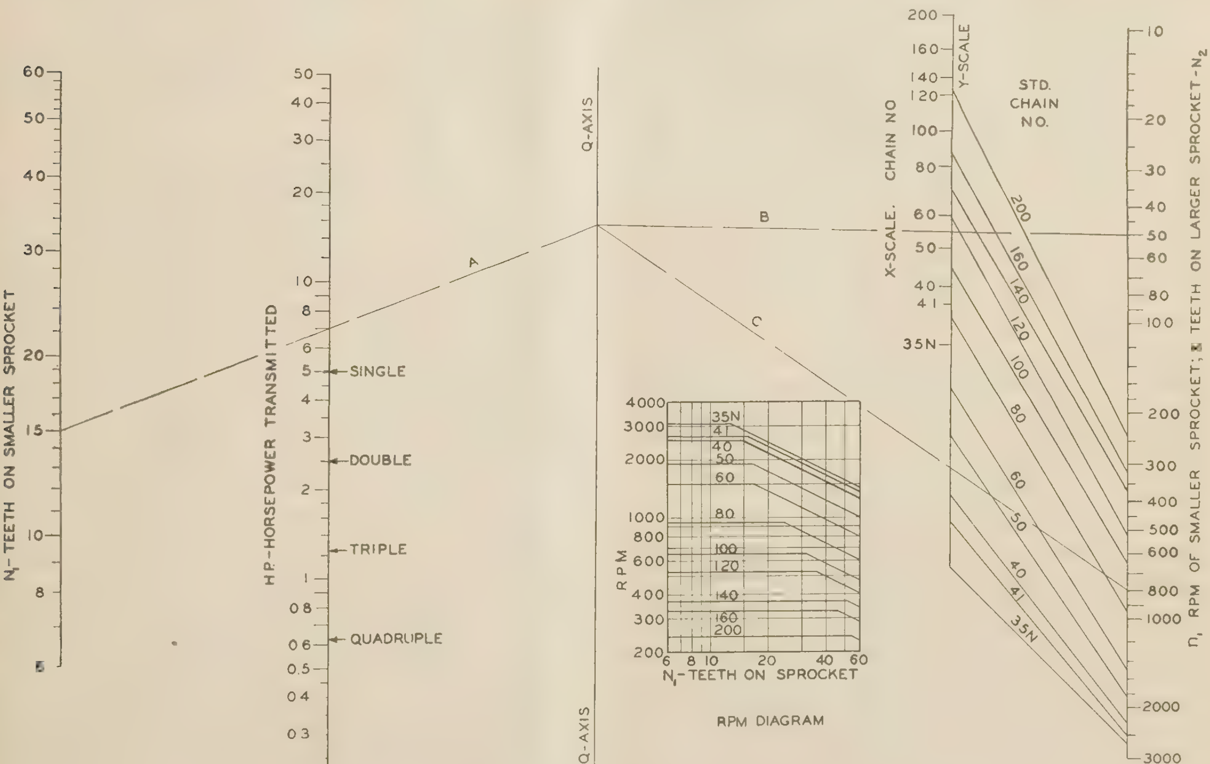

Fig. 1 Alignment Chart for Selecting Roller Chains

Operation: (1) Draw index A through N_1 and hp to the Q-axis, (2) draw index B from the Q-axis to N_2, (3) draw index C from the Q-axis to n_1. A chain is satisfactory if index C cuts the inclined line with that number, and if index B passes below the number on the X-scale, and if the allowable rpm for the number of teeth, N_1, as obtained from the rpm diagram, is not exceeded.

point on the Q-axis to n_1. Therefore, both are satisfied by chains in the inclined lines cut by the index line from the Q-axis to n_1.

As an example, let it be required to select a chain to transmit 7 hp over sprockets having 15 and 50 teeth, where the smaller wheel turns 800 rpm. (1) Draw the index line, A, from 15 on N_1 through 7 on the horsepower scale to the Q-axis. (2) Through this point on the Q-axis draw an index line, C, to $n_1 = 800$, and note that chains Nos. 50 60, and 80 (cut by this line), satisfy two of the three conditional equations. (3) The index line, B, from the Q-axis to $N_2 = 50$, cuts the X-scale below chains Nos. 60, 80, 100, etc., only two of which satisfy the first two conditions. Hence, either chain No. 60 or chain No. 80 is satisfactory for this example.

This chart as shown is for single chains only. If it is found that a single chain will not satisfy the conditions, the chart may be easily modified to select multiple chains on the same basis. If the horsepower scale is made movable along its axis, then to select double chains, the arrow at 5 on the horsepower scale is matched with the arrow marked "double," and the chart is used in the same manner as just described. The procedure for triple and quadruple chains is obvious. Incidentally, shifting the horsepower scale in this way has the effect of dividing by 2 for double chain, by 3 for triple chain, and by 4 for quadruple chain. Therefore, one can use the chart as it is, if the horsepower is divided by 2, 3, or 4 as the case may be, when selecting multiple chains.

The conditions limiting the rpm are not completely fulfilled by the alignment chart, therefore, after tentatively selecting a chain by means of the chart one should refer to the rpm diagram shown here, or Table 1 in the author's paper, to make sure that the allowable speed is not exceeded. It will be observed that the No. 60 chain found in the example is satisfactory for 1500 rpm unless the number of teeth is greater than 17; above 17 teeth a gradual reduction in speed must be made. No. 80 chain is satisfactory for 940 rpm if the number of teeth is less than 24, and above 24 teeth a gradual reduction must be made as shown by the inclined portion of the curve marked 80.

The time required to select a chain by means of the chart is seldom greater than a minute if N_1, N_2, n_1, and hp are known. Obviously, the chart may be used inversely to find the maximum horsepower which may be transmitted or the limiting numbers of teeth that satisfy the two equations and the set of rpm curves.

Author's Closure

The closure can be used to no better advantage than to summarize briefly the conclusions of the mathematical research advanced in the paper and to add appropriate remarks relative to the possibilities and limitations of the theory, the formulas, and the tables developed.

Maximum Revolutions per Minute and Chain Velocities. Table 1 of the paper should have been provided with heavy lines drawn so as to divide it into three areas: An upper left-hand area showing figures calculated from formulas [6] and [7]; an upper right-hand area calculated from formulas [8] and [9]; and a lower left-hand area calculated from formulas [4] and [5]. Each of these formulas contains a constant the value of which can be determined properly only from practical tests. Arbitrary values have been used in this table only to show the general plan. They are subject to change so as to conform to facts as they may be discovered from a study of actual performance in practice.

Formulas [8] and [9] affecting the upper right-hand area of the table are built upon a theory the only excuse for the existence of which is that it seems to satisfy, to an approximate extent, the general belief that centrifugal force in some way limits the allowable chain velocity. Not only are the constants in formulas [8] and [9] subject to change, but also the structure of the formulas may be changed in case a more plausible theory is discovered.

Merely changing the constants in formulas [4] to [9], inclusive, would not alter the theory on which they are based. If, for example, it is found that the figures for a 17-tooth sprocket carrying a No. 40 chain should be 10 per cent higher than those given in Table 1, the constants in formulas [8] and [9] can be increased 10 per cent and this would increase most of the figures in the upper right-hand area and cause a shift in the location of the line dividing that area from the lower left-hand area.

Conditions for a Uniform Chain Elongation. This mathematical research on the rapidity of chain wear has been conducted on the assumption that it would be desirable to know just how the life of a roller chain is affected, in theory at least, by changes in pitch, width, pin diameter, chain length, and the number of sprocket teeth. Disregarding centrifugal pull, the effect of which is almost negligible, it is concluded from formulas [20] and [21] that if uniform conditions of lubrication and service are assumed, the rate of chain elongation due to wear is proportional to

$$H(N_1 + N_2)/BPN_1N_2L_p$$

If now it is assumed that all chain drives are to be designed so that, under uniform conditions of lubrication and service, the chains will reach a 3 per cent elongation at the end of a specified number of hours of service, as 10,000 hours for example, the horsepower to be transmitted would have to be calculated from the formula

$$H = KBPN_1N_2L_p/(N_1 + N_2) \quad \text{[22a]}$$

where K is a constant to be determined by experiment. The constant K has been taken tentatively equal to 0.023 for the sake of illustrating the use of the formula and of tables computed from the formula. This formula and the ones given later in this closure are numbered as supplements to those given in the paper to facilitate reference to the paper.

In order to simplify the horsepower tables computed from formula [22a], it has been further assumed that the chain length L_p contains in every case as many links as there are teeth in the two sprockets. That is, the tabulated horsepowers in Table 3 in the paper are for chains containing exactly $(N_1 + N_2)$ links. This must not be overlooked since, when $L_p = N_1 + N_2$, the center distance is very much shorter than would be used in the average drive, and Table 3 is only designed to give horsepower ratings for drives having short center distances. In a drive using 16 and 25 teeth, $N_1N_2 = 400$ and the chain length is assumed as $16 + 25 = 41$. The center distance would then be about 10 pitches, and at 1200 fpm the rated horsepower is 7.3. If the chain length is doubled the center distance would be about 31 in., which is much closer to the average distance in actual practice. This makes the rated horsepower 14.6 instead of 7.3 as found in the Table 3.

If Tables 3 and 4 had been based upon an assumed minimum chain length of $2(N_1 + N_2)$ pitches, the constants in formulas [26], [28], and [30], inclusive, would be doubled, the horsepower ratings in the upper right-hand area would all be increased, and the heavy stepped line would be moved to a higher position.

Table for the Selection of a Suitable Chain. Table 5, given with this closure, illustrates an attempt to provide some means by which a proper chain can be quickly selected for a drive when the horsepower, the number of teeth N_1 and N_2, the revolutions per minute, and the approximate center distance (in pitches) are known. Disregarding the slight effect of centrifugal action, but taking account of the chain length, we have from formula [28]

$$H = 0.023PBN_1N_2\,L/(N_1 + N_2) \quad \text{[28a]}$$

TABLE 5 VALUES FOR THE SELECTION OF ROLLER CHAINS

Single-width chains					Triple-width chains				
X	Y	Z	Std. chain no.	Pitch and width	X	Y	Z	Std. chain no.	Pitch and width
2.46	0.185	2662	35N	$^3/_8 \times {^3/_{16}}$	7.41	0.555	2662	T35N	$^3/_8 \times {^3/_{16}}$
4.00	0.314	2500	41	$^1/_2 \times {^1/_4}$	12.05	0.942	2500	T41	$^1/_2 \times {^1/_4}$
4.96	0.427	2310	40	$^1/_2 \times {^5/_{16}}$	14.90	1.281	2310	T40	$^1/_2 \times {^5/_{16}}$
7.69	0.850	1830	50	$^5/_8 \times {^3/_8}$	23.10	2.550	1830	T50	$^5/_8 \times {^3/_8}$
11.90	1.534	1500	60	$^3/_4 \times {^1/_2}$	35.60	4.700	1500	T60	$^3/_4 \times {^1/_2}$
20.20	3.470	940	80	$1 \times {^5/_8}$	60.5	10.41	940	T80	$1 \times {^5/_8}$
30.50	6.320	645	100	$1^1/_4 \times {^3/_4}$	91.5	18.96	645	T100	$1^1/_4 \times {^3/_4}$
47.5	11.43	520	120	$1^1/_2 \times 1$	142.0	34.29	520	T120	$1^1/_2 \times 1$
57.8	16.00	370	140	$1^3/_4 \times 1$	172.0	48.00	370	T140	$1^3/_4 \times 1$
80.6	24.95	325	160	$2 \times 1^1/_4$	241.8	74.85	325	T160	$2 \times 1^1/_4$
122.0	51.26	240	200	$2^1/_2 \times 1^1/_2$	366.0	153.78	240	T200	$2^1/_2 \times 1^1/_2$
Double-width chains					Quadruple-width chains				
4.95	0.370	2662	D35N	$^3/_8 \times {^3/_{16}}$	9.90	0.740	2662	Q35N	$^3/_8 \times {^3/_{16}}$
8.06	0.628	2500	D41	$^1/_2 \times {^1/_4}$	16.10	1.256	2500	Q41	$^1/_2 \times {^1/_4}$
9.93	0.854	2310	D40	$^1/_2 \times {^5/_{16}}$	19.84	1.708	2310	Q40	$^1/_2 \times {^5/_{16}}$
15.40	1.700	1830	D50	$^5/_8 \times {^3/_8}$	30.80	3.400	1830	Q50	$^5/_8 \times {^3/_8}$
23.70	3.065	1500	D60	$^3/_4 \times {^1/_2}$	47.4	6.130	1500	Q60	$^3/_4 \times {^1/_2}$
40.30	6.940	940	D80	$1 \times {^5/_8}$	80.6	13.88	940	Q80	$1 \times {^5/_8}$
61.0	12.64	645	D100	$1^1/_4 \times {^3/_4}$	121.8	25.28	645	Q100	$1^1/_4 \times {^3/_4}$
95.0	22.86	520	D120	$1^1/_2 \times 1$	188.0	45.72	520	Q120	$1^1/_2 \times 1$
115.7	32.00	370	D140	$1^3/_4 \times 1$	231.5	64.00	370	Q140	$1^3/_4 \times 1$
161.0	49.90	325	D160	$2 \times 1^1/_4$	322.4	99.80	325	Q160	$2 \times 1^1/_4$
244.0	102.52	240	D200	$2^1/_2 \times 1^1/_2$	488.0	205.04	240	Q200	$2^1/_2 \times 1^1/_2$

Substituting $2C + [(N_1 + N_2)/2]$ for L in formula [28a], this substitution being sufficiently accurate, we obtain

$$H = 0.023PBN_1N_2[2C/(N_1 + N_2) + 1/2] \quad \ldots\ldots [28b]$$

$$H = 0.0127PBdN_1n_1 \quad \ldots\ldots\ldots [29b]$$

The horsepower rating can now be said to be the lesser of the two values computed from formulas [28b] and [29b]. The effect of a change in center distance is shown in the following example.

Let P = 1 in., N_1 = 15, N_2 = 21, C = 21, and the speed = 547 rpm. Then the rated horsepower is 10.5.

If the center distance is changed to 42 pitches, the rated horsepower is one and two-thirds times as great, or 17.5 hp.

Now from formulas [28b] and [29b]

$$23PB = \frac{1000H}{N_1N_2[2C/(N_1 + N_2) + 1/2]} \quad \ldots\ldots [28c]$$

$$12.7PBd = 1000H/N_1n_1 \quad \ldots\ldots\ldots [29c]$$

The first member of formulas [28c] and [29c] has a fixed value for each standard *chain*, and the second member has a particular value for each chain *drive*.

In this closure the author gives a table, numbered Table 5 as a continuation of the table numbers given in the paper. In Table 5, column X shows values of $23PB$ for the several standard chains listed in Table 1 of the paper. The column headed Y shows values of $12.7PBd$ for the same chains, and the column headed Z gives the max rpm for a 17-tooth sprocket as listed in Table 1 of the paper.

If we compute

$$x = \frac{1000H}{N_1N_2[2C/(N_1 + N_2) + 1/2]}$$

and

$$y = 1000H/N_1n_1$$

for a proposed drive, and z is taken as the revolutions per minute of the smaller sprocket, it is then only necessary to refer to the table for a set of values of X, Y, and Z, each of which is greater than x, y, and z. The standard chain number is then found in column 4 of Table 5. In most cases there will be several possible selections, and the one highest in the list will usually be the most economical. A quick reference to the table of maximum revolutions per minute will give an adequate check on the selection.

As an example, let it be required to select a chain to transmit 15 hp over sprockets having 15 and 25 teeth, the smaller wheel turning at 900 rpm, and the center distance to be about 30 pitches. Then

$$x = \frac{1000 \times 15}{15 \times 25\{[(2 \times 30)/(15 + 25)] + 0.5\}} = 20$$

$$y = 1000 \times 15/15 \times 900 = 1.111, \quad \text{and } z = n_1 = 900$$

In Table 5, sixth row from top, the values of X, Y, and Z are 20.20, 3.470, and 940, respectively, all of which are greater than the required values. The standard chain number is 80 and is a 1-in. pitch chain. It will be noticed that double-, triple-, and quadruple-width chains are listed as well as the single-width chain. If a selection cannot be made from the list of single chains, reference is made to the double-width chains, then to triple-width or the quadruple-width chains. In this case chain number D60 could be used but it would not be as economical as the single-width chain No. 80.

Referring to J. N. Arnold's discussion, the difference in the results obtained from his chart and those obtained by the method just described is due to the fact that the author is here taking account of the effect of a change in center distances, whereas Mr. Arnold's chart was designed to give results corresponding to formulas [28] and [29] and to Table 3. Doubtless Mr. Arnold's chart could easily be modified to take care of varying center distances if desired.

Effect of an Excessive Number of Teeth. One more comment seems to be important. An inspection of Table 4 of the paper shows that in the case of drives having over 60 teeth in the larger sprocket the horsepower ratings seem to run higher than they should. The reason for this is that the present research has dealt only with the rate of chain elongation and has not taken into account the fact that, whereas chains operating over sprockets with less than 60 teeth can often stretch 3 per cent before being discarded, this is not true where the sprockets have considerably more than 60 teeth.

In the case of 90 teeth the chain will climb very nearly to the top of the teeth when the elongation is only 2 per cent; and this means that the allowable life of the chain on a 90-tooth sprocket is only two-thirds as great as one running on a 60-tooth sprocket. To take account of this a note may be attached to Table 5 stating that, where N_2 is more than 60, the values of x are to be multiplied by $N_2/60$ before being applied to the table.

The Leakage of Steam Through Labyrinth Seals[1]

R. L. Scorah.[2] The subject of flow through labyrinth seals has received rather limited study, even though it has been known for sometime that the older formulas do not always give reliable results. Mr. Egli's paper will no doubt form a welcome addition to the literature of this field.

It should be pointed out, however, that the static tests referred to in this paper do not simulate the real conditions of operation. In practice, the leakage steam flows in a different sort of channel, one part of which is stationary while another part, the shaft, rotates at high speed. It has not been demonstrated that the leakage is independent of the speed of rotation. Furthermore,

[1] Published as paper FSP-57-5, by Adolf Egli, in the April, 1935, issue of the A.S.M.E. Transactions.

[2] Mechanical Engineering Department, Stanford University, California.

the argument presented here is confined to the gas phase of the flowing fluid, that is, superheated steam. In some cases, the steam entering the seal will be wet or only slightly superheated. Under these conditions, the effect of supersaturation and water drops further complicate the problem. In view of the wide variety of designs used in practice and the absence of precise information regarding supersaturation and the behavior of entrained water drops, it would seem expedient at this time to investigate experimentally the leakage through various labyrinth constructions using a rotating-shaft element. Such experiments would not only provide reliable leakage data for the particular constructions tested but would also serve as a helpful check on the rational theoretical treatment of the problem.

H. G. Yates.[3] The writer is interested in knowing if Mr. Egli has studied the effect of inclining the baffle strips at an angle of 45 deg. Glands of this type in conjunction with a spring mounting have been found satisfactory. It is uncertain, however, how far their good sealing properties are due to the inclination and how far to the accurate maintenance of fine clearances by means of the spring mounting. Any gain in sealing properties would probably be proportionately smaller with very fine clearances (of the order of 0.01 in. or less) as the small radius, which necessarily exists on even a "sharp" baffle strip, would assume proportionately greater importance, and the constriction would cease to be a sharp-edged orifice and would tend toward one with well-rounded edges. It is presumably this effect which gives rise to the curves of Fig. 18 of the paper. An important advantage of the inclined baffle strip is that in the event of local heating due to a momentary rub, the strips tend to cockle up away from the shaft rather than to expand toward it.

Another gland which has been employed frequently is of the "straight-through" type but with the baffle strips adjacent to a number of thin closely pitched ribs turned on the shaft or gland sleeve, of different pitch from the strips themselves, so that there is no fear of a straight blow-through if the rotor should be moved in an axial direction. Besides having the property of rapid cooling in the event of a rub, it is believed that the "valleys" between the ribs or pips serve to break up the steam flow and give almost as good a seal as a gland of the well-staggered type. The writer would welcome Mr. Egli's opinion on this.

B. Hodkinson.[4] Mr. Egli refers more than once in his paper to the high velocity through the last throttling and points out that the Fanno line may at this stage be departed from.

A way of allowing for this in the calculation suggests itself. The pressure before the last throttling may be taken as twice the final pressure, and the calculation carried only to this point. In other words, the last throttling could be done away with, and the final pressure doubled. This would raise the calculated figure slightly, except in odd cases, and in the light of some tests at Trafford Park, Manchester, on a grooved valve spindle, would make the answer more reasonable, because we found a greater mass flow than appeared, according to calculation, to be possible.

The odd cases where the calculated discharge would not be increased occur when p_2/p_1 is higher than about 0.2, and then the last throttling probably does not carry sound velocity. Most practical labyrinths operate with p_2/p_1 a good deal less than 0.2. Thus, it would be well to use the above suggested way only when p_2/p_1 equals, say, 0.1 or less.

[3] English Electric Co., Ltd., Rugby, England.

[4] Turbine Experimental Section, Mechanical Engineering Department, Metropolitan Vickers Electrical Co., Ltd., Manchester, England.

Author's Closure

The author fully realizes the fact, mentioned by Prof. R. L. Scorah, that the actual flow conditions are not correctly simulated in the static tests referred to in the paper. Some of Friedrich's[5] tests, which were made with a rotating shaft, show the effect of peripheral speed to be rather small. At 200 ft per sec the decrease of leakage due to the speed of the shaft has been found to be 3.4 per cent. For moderate peripheral speeds, therefore, the static tests will not be much in error.

There is a further point with a view to simulating the actual flow conditions, which has been neglected in the static tests. It is the exact shape of the edges of the sealing strips after they have been rubbing in the running turbine. The strip generally becomes burred over and the sealing edge is no longer sharp, as assumed in the static tests, but is in cross-section shaped rather like a mushroom. The characteristic function ψ of such a mushroom-shaped throttling undoubtedly is somewhat different from the function ψ of a sharp-edged orifice shown in Fig. 2 of the paper. It would be principally possible to determine this function by testing a single "mushroom-shaped" strip. Curves φ versus pn/po similar to those of Fig. 7 of the paper could then be constructed with the methods there presented. For most practical purposes, however, the φ curves of Fig. 7 will be sufficiently accurate.

The author has had no experience with the sealing properties of the particular types of baffles described by H. G. Yates, and does not believe that by inclining the labyrinth strips the amount of leakage for a given clearance should descrease measurably. If such glands have proved successful in the turbine, it apparently is due to the maintenance of a fine clearance. The "straight-through" type seal with strips adjacent to a mumber of closely pitched ribs is probably somewhat tighter than the "straight-through" type gland referred to in the paper. It is his opinion, however, that this type of packing will not reach the good sealing properties of a well-staggered labyrinth. The leakage jet issuing from one clearance very easily "bridges" over the narrow "valleys" between the ribs by simply maintaining a series of stationary vortices in each valley.

The method of considering the great pressure drop across the last throttling, as suggested by H. B. Hodkinson may give fairly correct results, although the assumption of the pressure before the last throttling to be twice the final pressure is quite arbitrary. There is, however, no need of treating separately the last throttling when following the methods outlined in the paper. The curves of Fig. 7 and 7*a* automatically take into account the flow characteristics of each throttling in the labyrinth including the last one.

The physical nature of the leakage flow along a grooved valve stem, to which Mr. Hodkinson refers, is quite different from that in a labyrinth packing. Here the loss of kinetic energy due to friction in the narrow space between stem and bushing must be taken into consideration whereas in a labyrinth the friction in the short passage through the throttling gap plays a minor rôle only.

[5] "Untersuchungen ueber das Verhalten der Schaufelspaltdichtungen in Gegenlauf-Dampfturbinen," by H. Friedrich, *Mitt. Forsch. Anst. G. H. H.*, Oct., 1933.

FSP-57-14

Cause and Prevention of Turbine-Blade Deposits[1]

By FREDERICK G. STRAUB,[2] URBANA, ILL.

A survey was made of power plants encountering turbine-blade deposits and it was found that this trouble results from contamination of the steam by the boiler water. A study in the laboratory showed that sodium hydroxide is the material which causes the sticking to the blades. It was also shown that when the sodium hydroxide is neutralized and changed to a salt such as sodium carbonate, it will not adhere. The presence of sufficient amounts of inert salts, such as sodium sulphate, with the sodium hydroxide also will stop the deposit from forming.

The action of sodium sulphate in preventing the blade deposits was studied in a large central power plant, and the deposit was materially reduced. A small testing unit for detecting the presence of adhering salts in the steam was developed and used in the power-plant tests.

STEAM-electrical generating stations have encountered difficulty in the form of fouling of turbine blades. This difficulty has become of major importance in many large stations whereas it has only meant annoyance in other stations.

There are several types of deposits which form on the turbine blading and cause this fouling. One is that which is apparently caused by a deposition of solids carried in the steam from the boiler water and another is that caused by a chemical reaction between chemicals in the steam and the material in the turbine blade. The first type is most common and is readily distinguished because it is largely water-soluble and is washed off with comparative ease, whereas the other type adheres tenaciously to the blades.

The difficulty caused by the deposition of solids carried in the steam appears to be the one causing the major difficulty. The efforts of this research have been expended entirely toward the study of this type of deposit and no study has been made of the other type.

Industrial Experience

A survey was made of power plants encountering this trouble as well as those not having difficulty with blade deposits.

The summary of conclusions reached after assembling the available data in regard to turbine-blade deposits is as follows:

(1) The deposits formed independent of steam pressure or temperature.

(2) The deposits were not proportional to the total concentration of solids in the boiler water.

(3) The total solids in the steam were very low in many plants where deposits occurred.

(4) The deposits all contained sodium present as hydroxide, carbonate, or silicate.

(5) The deposits were practically all water-soluble.

LABORATORY EXPERIMENTS

Possible Cause of Deposits

The deposits which were formed in the plants under consideration all originated in the boiler water. A small amount of the boiler water may be easily carried into the steam. Actually, this amount does not have to be more than one-tenth of one per cent of moisture in the steam to cause appreciable trouble at the turbine. Thus if a boiler water should contain about 300 ppm total solids, which is a rather low concentration, and one-tenth of one per cent of the steam were present as boiler water, the steam would contain 0.3 ppm of total solids. This appears a negligible amount; however, with a turbine using one million pounds of steam per hour, 8 lb of solids would pass through the turbine in 24 hr. If only 10 per cent of this were to adhere to the turbine blades, the efficiency and capacity loss would increase at an alarming rate. Thus if there is present in the boiler water any material which would cause adherence to the turbine blades, the better quality steam might cause appreciable difficulty.

In order to determine whether there is present in the average boiler water material which would cause adherence or sticking to the turbine blade a study was made of the behavior of the various salts encountered in boiler waters as they pass from solution in wet steam to superheated steam. If only pure water were present in the boiler and a small amount of the boiler water were mechanically carried into the steam and to the superheater, the droplets of water would vaporize in the superheater and no free moisture would be present in the superheated steam. However, if sodium chloride, sodium carbonate, or sodium sulphate were present in the boiler water the droplet of water entering the superheater with the steam would contain a dilute solution of the salt or salts. As the steam became superheated the water would vaporize and leave a dry salt or powder of the salt or salts with the superheated steam. Consequently, they would pass through the steam pipes and turbine as a fine powder or dust and cause no appreciable difficulty. This is based on the assumption that there is only a small percentage of moisture in the steam. However, if sodium hydroxide were present this would not happen. This results from the fact that the behavior of sodium-hydroxide solutions is entirely different from that of the other salts discussed.

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[1] Part of the research conducted in cooperation with the Utilities Research Commission, Chicago, Ill. Published by permission of Dean M. L. Enger, Director, Engineering Experiment Station, University of Illinois.

[2] Special Research Assistant Professor in Chemical Engineering, University of Illinois. Mr. Straub was graduated from the University of Illinois in 1920. After leaving the university, he was associated with Mellon Institute, Pittsburgh, Pa.; Semet Solvay, Syracuse, N. Y.; and Guggenheim Brothers Research Laboratories, New York, N. Y. He holds the degrees of Master of Science and Metallurgical Engineer from Pennsylvania State College. He has been conducting special research for the Utilities Research Commission, Inc., on boiler-feedwater treatment for the last eleven years at the University of Illinois. This has included work on determining the causes and methods of prevention of embrittlement in steam boilers and a study of the methods of preventing scale in high-pressure boilers.

Contributed by the Power Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers to be held in New York, N. Y., December 2 to 6, 1935.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication in a later issue of Transactions.

When a solution of sodium chloride, carbonate, or sulphate is boiled, the solution is concentrated as the steam is released and the solution concentrates until a saturated solution is obtained. Any further release of steam results in the precipitation of the salt. Eventually, if the boiling is continued, all the water is evaporated and dry salt or salts are left behind. If a solution of sodium hydroxide is boiled, the solution concentrates with a continual increase in temperature until a concentration is reached where the concentrated solution is in equilibrium with the vapor pressure of the surrounding atmosphere and further heating will cause a release of steam to the surrounding atmosphere with an increase in concentration of the caustic solution and an increase

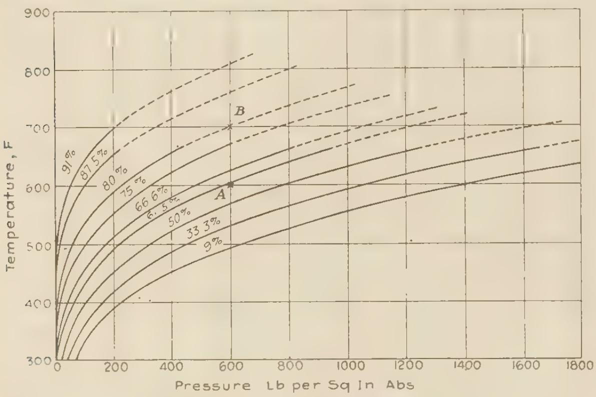


Fig. 1 Concentration of NaOH in Relation to Steam Pressure and Temperature

in the temperature of the solution. This is shown graphically in Fig. 1 and may be illustrated as follows:

If a drop of water containing sodium hydroxide in solution leaves a boiler at a pressure of 600 lb per sq in. abs and enters the superheater with the steam, the water will vaporize, thus concentrating the solution. When the temperature of the steam reaches 600 F the concentration of the caustic will represent about a 60 per cent solution, shown as *A* in Fig. 1. If the temperature and pressure remained constant, the droplet of caustic would remain 60 per cent sodium hydroxide and 40 per cent moisture, even in contact with superheated steam. As the temperature increases, the pressure remaining constant, the caustic concentration will increase. At a pressure of 600 lb and a temperature of 700 F, the concentration would be as shown at point *B*, an 80 per cent solution.

The concentration of caustic in equilibrium with superheated steam at various pressures and temperatures is shown in Fig. 1. This shows that the concentrations of sodium hydroxide reached in the average higher pressure plant is between 80 and 90 per cent. A solution of sodium hydroxide containing between 10 and 20 per cent moisture at these temperatures will be in a pasty or semifluid state and in going through the turbine will adhere to the blades. The other salts, present as a fine dust, will adhere with the sodium hydroxide, but not necessarily in the relative proportions in which they exist in the boiler water. When the temperature of the steam in the turbine is lowered until saturated steam exists, these salts will be dissolved and thus wash off of the blades. The experience which turbine operators have had with washing of turbines confirms this.

In order to substantiate this theory tests were run in the laboratory. The procedure followed in these tests was to produce steam in a small boiler, contaminate the steam with a solution of a desired salt or combination of salts, superheat the contaminated steam, pass the steam through an orifice so that it impinged on a stationary blade, and then condense the steam at a predetermined pressure.

Fig. 2 shows the apparatus used for the tests. Condensed steam from the steam-heating system was passed through a small deaerating heater and then pumped to an electrically heated boiler to generate steam. The steam was then passed under pressure into the contaminator. The contaminator contained a dilute solution of the contaminating salt or salts and was so built that the steam entered the bottom, bubbled up through the solution, and was taken off at the top. The contaminator was heated by means of a hot plate to compensate for radiation loss and to maintain a constant level of solution. The steam leaving the contaminator was passed through an electrically heated superheater, then through an orifice so as to impinge on the removable blade, and then through a copper-coil condenser to the condensate storage. The steam pressure and the superheated-steam temperature were maintained constant by means of potentiometer temperature regulators. The temperature of the blade on the side opposite the point of steam impingement was recorded on a recording potentiometer temperature recorder. When a vacuum was maintained in the condenser this was done by evacuating the condensate storage to the desired pressure. The pressure in the container holding the blade was determined by means of a pressure gage.

If the temperature and pressure existing in the generating

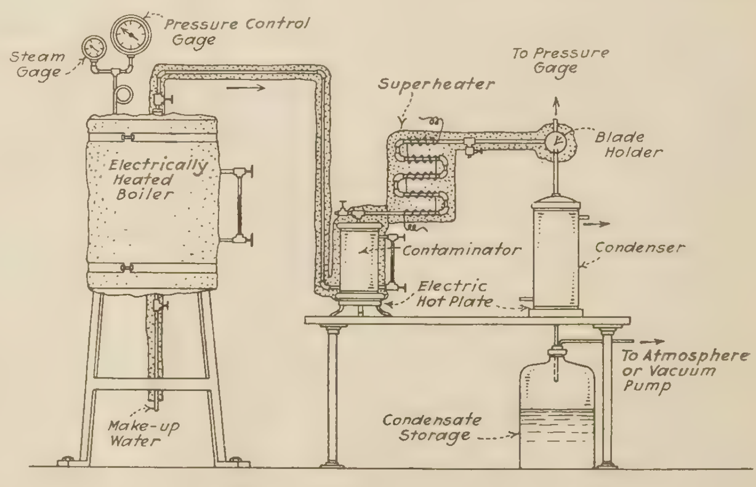


Fig. 2 Assembly of Apparatus

stations was in range of 600 lb and 700 F, a concentration of about 80 per cent sodium hydroxide would result. In order to simplify the construction of the test apparatus the tests were run at 45 lb per sq in. abs and 400 to 700 F, with a pressure of 0.8 lb in the blade chamber. Reference to the curve in Fig. 1 will show that under these conditions the sodium hydroxide should reach a concentration of between 85 and 90 per cent. This concentration is in the range of that reached in turbines. The orifice was $^1/_{16}$ in. in diameter and the blade was set at an angle of 30 deg to the line of steam flow leaving the orifice.

The first tests were run for 22 hr with 2 grams sodium hydroxide added to the contaminator and the temperature of the superheated steam varied between 400 and 700 F. Deposits formed on the blades. The theoretical composition of the sodium hy-

droxide in the steam at the time of contact with the blade during these tests varied from about 85 per cent at 400 F to about 98 per cent at 700 F. Tests run with distilled water alone in the contaminator gave clean blades. When sodium chloride was added to the contaminator and tests run at 500 F, no deposit formed on the blade in the area of high velocity. Similar tests with sodium sulphate gave like results. Trisodium phosphate and sodium silicate both formed deposits on the blade.

The sodium-hydroxide, silicate, and phosphate deposits were formed because hydroxide was present in all these salts. If this is true, any chemical added to the contaminated steam which would neutralize the hydroxide should prevent the hydroxide from forming. In order to prove this hypothesis the apparatus was modified so that carbon dioxide could be slowly added to the steam beyond the contaminator and ahead of the superheater. The carbon dioxide, if present in excess, should react with the hydroxide to form sodium carbonate. This salt should exist as a dry powder in the steam and no deposit should form. When carbon dioxide was added after contamination with sodium hydroxide, sodium silicate, and sodium phosphate, no deposits formed. This showed conclusively that the sodium hydroxide was the binding or adhering agent, and if it were changed to carbonate prior to reaching the blade, no deposit would form.

In order to study the possibility of the utilization of carbon-dioxide treatment the apparatus was modified so that the carbon dioxide could be added to the superheated steam instead of the wet steam. When tests were run using this method of treatment the deposits continued to form even when large excesses of carbon dioxide were present.

This change in the action of the carbon dioxide is easily explained. When the gas is added to the wet steam the sodium hydroxide in the droplets of water is in a dilute solution. In this state the carbon dioxide also is soluble in the water depending upon its partial pressure in the steam. As the two chemicals are in dilute solutions, they are strongly ionized and will react readily so that the sodium carbonate is formed before the steam is superheated. However, when the steam is superheated before adding the carbon dioxide the excess water present is converted to steam and the sodium hydroxide is present in the form of a highly concentrated solution, more than 80 per cent NaOH. In such a solution the hydroxide is only slightly ionized, the carbon dioxide is present in the steam as a dry gas, consequently there is practically no reaction between the gas and the sodium hydroxide. This appeared to make the carbon-dioxide method of treatment rather complicated.

Concentration of Caustic Necessary for Deposit

After this substantiation of the hypothesis that sodium hydroxide is the basic adhering material in the turbine-blade deposits it seemed desirable to obtain further data on the limiting concentration of sodium hydroxide in the steam which would cause blade deposition and on other methods of prevention of this troublesome deposit.

In the apparatus already described the carry over or contamination of the steam was excessive in the early period of the test. This was shown by the fact that with a constant water level in the contaminator the concentration of the contaminating salts steadily decreased. At the same time it was almost impossible to know the amount of contamination in the steam at any particular time.

In order to overcome these objections the apparatus was redesigned as shown in Fig. 3. The blades were made from watch springs ($^3/_8$ in. by 0.020 in.) and were removed at the end of each test. The steam nozzle was made by slotting the end of the tube

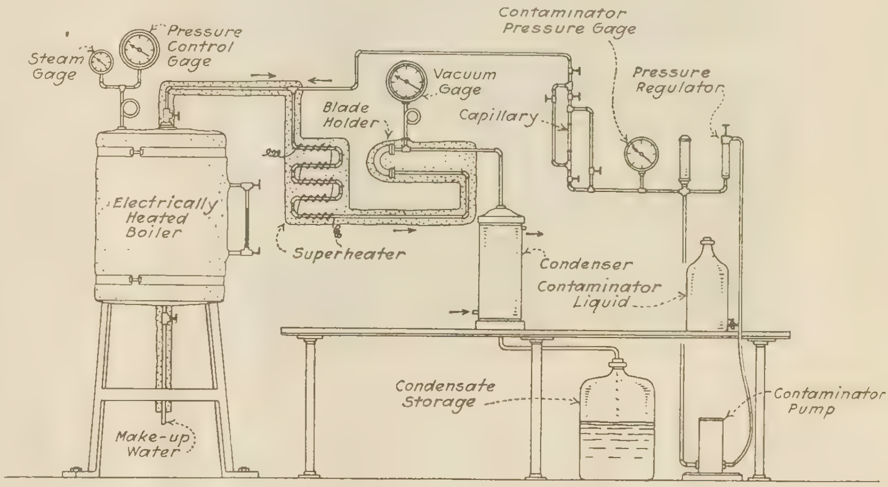


Fig. 3 Assembly of Apparatus, Redesigned

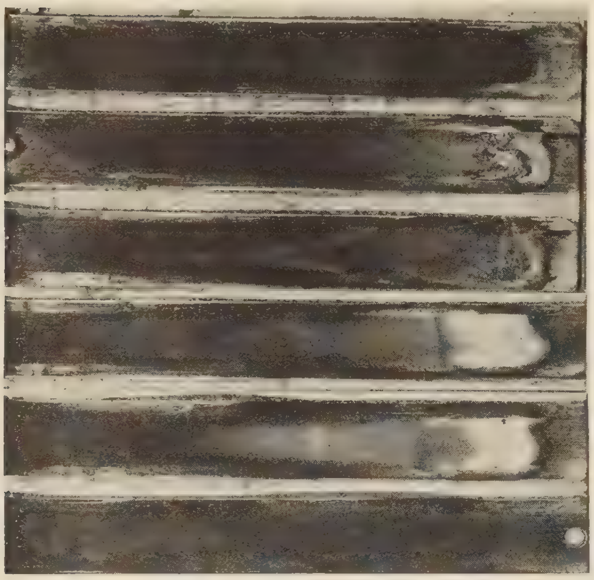
and this brought a sheet of steam in contact with a curved blade in a manner similar to that in which the steam strikes the blade of a turbine.

The contamination was brought about by pumping the solution containing the desired concentration of chemicals into a small reservoir where it was held at a definite pressure by means of a ball-seat pressure regulator. The excess liquid leaving the regulator was returned to the pump suction. The reservoir was connected through a small capillary orifice to the steam line ahead of the superheater. The steam pressure was maintained constant by means of a steam pressure regulator connected through relays to electrical heat controls on the boiler.

With a constant pressure drop across the orifice the amount of contaminating liquid entering the steam through the orifice was constant and depended upon the pressure drop across the orifice which was under definite control. About 0.55 $\pm$ 0.02 lb per lb per hour of contaminating liquid was fed into the steam. The total amount of steam—steam from boiler plus steam from contaminator liquid vaporized in the superheater—passing through the nozzle was about 10 lb per hr.

The water fed to the boiler was condensed steam from the steam-heating system which had passed through an electrically heated boiling deaerating heater prior to being pumped to the boiler. The boiler pressure was maintained at 40 lb gage. Several pressures in the chamber holding the blade were used.

Tests were then run using the new apparatus and holding the vacuum in the blade chamber at 20 in. of mercury. With the sodium hydroxide in the steam (calculated) at 48, 18, 5, 2, and 1.5 ppm deposits formed in 20-hr tests. Tests in which distilled water was added to the steam gave no deposit. When the steam was passed through the nozzle there was a marked change in velocity. The calculated velocity in the steam line ahead of the nozzle was 21.5 fps. The velocity through the nozzle was 1175



TEST NO.	CHEMICALS ADDED TO STEAM, PPM NaOH	Na_2SO_4	SiO_2
138	1.54	11.3	—
139	1.67	—	—
140	1.74	—	—
149	1.4	—	0.87
151	5.0	—	3.1
152	4.4	17.5	2.5

Fig. 4 Photographs of Blades After Tests

fps, and the velocity past the blade was 950 fps. Thus the velocity at point of contact with the blade was approximately 1000 fps.

The maintenance of a vacuum in the blade holder complicated the operation of the test. Consequently, tests were run in which atmospheric pressure was maintained in the blade holder. With such a change in conditions the velocity of the steam in the holder dropped from a nozzle velocity of 1175 fps to 320 fps. Undoubtedly the velocity at the point of impact with the blade was still in the range of 1000 fps. Tests run under these conditions with 4, 5, 4.0, 3.2, and 2 ppm of sodium hydroxide added to the steam gave deposits in the 20-hr tests. This showed that a deposit would form with the sodium-hydroxide content of 2 ppm in 20 hr. The remainder of the tests were run at this pressure unless otherwise noted.

Sodium sulphate alone was added so as to have 10 ppm in the steam and no deposit formed. A mixture of sodium hydroxide and sulphate was then added so that the NaOH was 4.2 and the Na_2SO_4 8.6 ppm in the steam and small deposits formed, but not as much as that formed by the sodium hydroxide in the absence of the sodium sulphate. The sodium hydroxide was then increased to 5.8 and the sulphate to 25 ppm and no deposit formed. This test was followed by one with a sodium-hydroxide concentration of 2.9 ppm and no sulphate. A definite deposit formed. This appeared to indicate that the sulphate was preventing the hydroxide from adhering to the blade.

Tests were then run with the sodium-hydroxide concentration held constant at 3 ppm and the temperature at 350, 400, and 615 F. A deposit formed in all these tests, indicating that changing the temperature did not have much effect upon the tendency of the sodium hydroxide to adhere to the blade.

The theoretical composition of the sodium hydroxide in the steam leaving the nozzle and in the blade holder are given in Table 1 for these temperatures. This shows that the caustic soda is adhering over a range of approximately 50 to 93 per cent and possibly a greater range, and illustrates the fact that temperature does not materially affect the tendency to deposit if the steam is superheated.

The procedure followed in running these tests involved attaching the blade holder after the steam was superheated and removing it while the steam was still superheated. Modification of this procedure so as to attach the blade holder before the steam was superheated but while the caustic solution was being pumped into the steam gave a condensate having a distinct caustic alkalinity which corresponded with the theoretical contamination. However, when the steam was superheated the condensate was practically free from sodium hydroxide. This showed clearly that the sodium hydroxide was being removed prior to the condensing of the steam. The deposit on the blade showed that the last bit of sodium hydroxide was being removed there.

It was doubtful that all the sodium hydroxide was being removed at the blade since a large amount was undoubtedly being removed in the superheater and piping ahead of the nozzle. At the end of each run the wet steam was run through the superheater and pipe after the blade holder had been removed. This contained appreciable amounts of caustic soda and proved that it was depositing in the superheater.

In order to determine the actual amount of sodium hydroxide in the steam at the nozzle and to see how completely it was being removed at the blade the apparatus was redesigned so that the conductance and pH value of samples of the contaminated steam and condensed steam before and after the blade holder could be determined. The boiler, superheater, blade holder, and condenser were the same as those previously used. A steam sampler was installed just ahead of the nozzle. The $^3/_8$-in. steam pipe was expanded into a $1^1/_4$-in. pipe in which two $^1/_8$ in. tubes had been inserted. These were drilled with small holes. The $^1/_8$-in.

TABLE 1 THEORETICAL CONCENTRATION OF SODIUM HYDROXIDE IN STEAM BEFORE AND AFTER NOZZLE

Steam press. gage, lb per sq in.	Steam temp, F	Pressure in blade holder	NaOH in steam: Before nozzle, per cent	NaOH in steam: In blade holder, per cent
40	350	atm	45	63
40	400	atm	62	75
40	500	atm	79	87
40	600	atm	90	93

tubes passed through condensers to a valve on the cool side of the condenser. In this manner a small amount of the contaminated superheated steam was removed continuously from the steam,

condensed under pressure, cooled, removed, and passed, without exposure to the air, to a conductivity cell and a glass electrode. Thus the conductance and the pH value of the condensed contaminated steam were determined. The condensed steam leaving the condenser after the steam had passed over the blade was also passed through a conductivity cell and a glass electrode. A comparison could thus be made of the values obtained from the condensed steam.

Distilled water was used for make-up. It was passed through the electrically heated boiling deaerating heater to remove dissolved gases such as oxygen and nitrogen and was pumped to the boiler. With this type of boiler feedwater the condensed steam from the boiler had a conductance of 0.05×10^{-6} mho or less.

When feedwater of this better quality was used the test procedure was modified so that the larger blade was weighed before and after the tests. The time of the tests was reduced from 20 to $5^1/_2$ hr. Tests for carbon dioxide were also run on the condensed steam after the blade in some cases.

Tests were run under these new conditions. Fig. 4 shows photographs taken of the blades after the tests. The blades have been flattened out. These results are interesting because they show that when sodium hydroxide is added to the steam in amounts as low as 0.55 ppm appreciable deposits, 2.7 mg in $5^1/_2$ hr, formed. This concentration in the steam is a maximum value.

The results in Table 2 also show that sodium sulphate and disodium phosphate did not cause deposits to form, whereas trisodium phosphate and sodium chloride caused small deposits to form.

TABLE 2 RESULTS OF TESTS USING STEAM CONTAMINATED WITH SULPHATE, CHLORIDE, AND PHOSPHATE

(Steam pressure 40 lb gage; steam temperature 500 F; pressure in blade holder, atmospheric; conductance of boiler feedwater, between 0.33 and 0.50×10^{-6} mho)

Test no.	Chemical contamination in steam ahead of superheater, ppm	Specific conductance $\times 10^{-6}$ mho, condensed steam samples: Before blade		After blade	Increase in weight of blade, mg	CO_2 after blade, ppm
137*B*	None	1.00	0.91	0.83	0.0	...
156	5.9 Na_2SO_4	3.72	3.67	2.38	0.2	0.9
158	5.5 NaCl	2.15	2.21	1.42	1.5	0.4
160	1.3 NaH_2PO_4	1.68	1.38	1.33	0.0	0.4
159	5.4 Na_3PO_4	6.90	5.95	3.38	1.1	1.0

Prevention of Deposit

Table 3 shows the results of tests run with varying amounts of sodium sulphate when the sodium-hydroxide content was constant. When the ratio of the sodium sulphate to the sodium hydroxide was 4.4 and greater the deposit did not form. This checked with the previous results and indicated that the presence of a sufficient amount of sodium sulphate with the sodium hydroxide prevents the formation of a deposit on the blade.

The addition of sodium chloride to the sodium sulphate so that the ratio of sodium sulphate to sodium hydroxide was 3.3 and sodium chloride to sodium hydroxide was 2.5, or the ratio of the total of the sodium sulphate and chloride to the sodium hydroxide was 5.8, did not entirely prevent the deposit; however, the deposit was less than it would have been with the sulphate alone present with the hydroxide, indicating that sodium chloride when present with the sulphate aids in the prevention of the deposit.

The presence of silicate along with the hydroxide tended to decrease formation of the deposit. When the sodium sulphate was present in an amount four times the sodium hydroxide, only a small deposit formed. This indicated that the sulphate was also effective in the presence of the silicate.

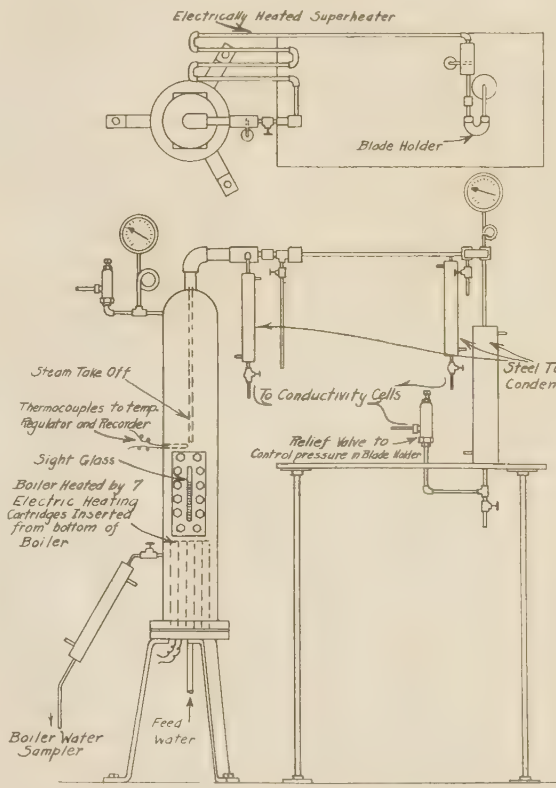


Fig. 5 Apparatus Used for Higher-Pressure Tests and Contamination of Steam From Boiler Water

Effect of Contaminating the Steam With Boiler Water

In the tests previously run the contamination was accomplished by adding the desired chemicals to the steam. This showed that if these chemicals were present in the steam the deposits could be formed or prevented. However, it seemed advisable to determine the actual ratios of the inhibiting chemicals necessary in

TABLE 3 RESULTS OF TESTS USING STEAM CONTAMINATED WITH SODIUM HYDROXIDE, SULPHATE, AND CHLORIDE

(Steam pressure 40 lb gage; steam temperature 500 F; pressure in blade holder, atmospheric; conductance of boiler feedwater, between 0.33 and 0.50×10^{-6} mho)

Test no.	Chemical contamination in steam ahead of superheater: NaOH, ppm	Na_2SO_4, ppm	NaCl, ppm	Ratio: $\frac{Na_2SO_4}{NaOH}$	$\frac{NaCl}{NaOH}$	Specific conductance $\times 10^{-6}$ mho, condensed steam samples: Before blade		After blade	Increase in weight of blade, mg	CO_2 after blade, ppm	pH: Before blade	After blade
139	1.67	...	..	0.0	...	1.70	1.89	0.88	5.2	...	...	6.3
145	1.50	5.5	..	3.66	...	1.87	2.28	1.17	1.5	1.3	...	6.3
147	1.50	6.6	..	4.4	...	2.14	2.15	1.53	0.3	...	...	6.3
150	1.23	5.2	..	4.2	...	1.75	1.93	1.47	1.6	...	...	6.2
214	1.85	9.25	..	5.0	...	....	4.13	3.18	0.2	...	8.4[1]	8.5[1]
138	1.54	11.3	..	7.3	...	4.65	4.55	3.03	0.0	...	...	6.7
148	1.50	5.0	3.7	3.3	2.5	1.23	1.29	1.24	0.5	0.7	...	6.2
146	1.6	...	3.0	..	1.8	1.52	1.86	1.00	5.4	0.7	...	6.2
215	1.80	9.0[2]	1.80	5.0	1.0	....	5.30	4.03	0.1	...	8.3[1]	8.4[1]

[1] Using steel condenser, others using copper condenser.
[2] PO_4 also present $1.1 \times NaOH$.

TABLE 4 RESULTS OF TESTS USING STEAM CONTAMINATED WITH BOILER WATER

(Steam pressure 600 lb gage; steam temperature 700 F; pressure in blade holder 500 lb gage; conductance of boiler feedwater, between 0.33 and 0.50 $\times 10^{-6}$ mho)

Test no.	Concentration of salts in boiler water: NaOH, ppm	NaCl, ppm	Na_2SO_4, ppm	Ratio: Na_2SO_4/NaOH	NaCl/NaOH	Specific conductance $\times 10^{-6}$ mho, condensed steam samples: Before blade	After blade	Increase in weight of blade, mg	pH[1]: Before blade	After blade
259	165	...	...	....	...	3.33	1.11	12.4	8.7	6.4
260	73	...	...	...	...	3.07	0.91	7.3	8.5	6.8
264	100	...	437	4.37	...	2.00	1.21	0.5	6.8	6.5
268	120	100	632	5.25	0.83	1.43	1.05	−0.1	6.6	6.7
269	260	72	1360	5.23	0.28	1.18	0.95	0.1	6.6	6.7
270	170	...	...	...	...	2.50	0.91	13.7	8.3	7.0

[1] Steel condensers.

the boiler water to prevent deposits. In such tests the contamination of the steam would result entirely from carry over from the boiler and the results would be more readily applied to actual operation.

The apparatus used for these tests is shown in Fig. 5. The superheater and the equipment beyond the boiler involved the same type of equipment as used before, except high-pressure fittings were used and stainless-steel pipe (18–8) was used for the superheater and subsequent piping. The boiler involved the use of seven 1.1-kw electric heating cartridges. These were inserted from the bottom into tubes welded into the bottom of the boiler. The water level and the behavior of the solution during boiling was observed by means of sight glasses placed on opposite sides of the boiler. The heat imput was controlled by means of potentiometer temperature regulators so as to maintain constant temperature and pressure in the boiler. The boiler feedwater was the same as previously described. The chemical contamination of the boiler water was accomplished by pumping the desired amount of chemical into the boiler at the beginning of the tests. Samples of the boiler water were taken for analysis at the beginning and end of each test. Condensed steam samples were taken ahead of the superheater, ahead of the nozzle and beyond the blade holder and the conductance and pH values were determined. These tests were run for $5^1/_2$ hr at a steam pressure of 600 lb per sq in. and a total steam temperature of 700 F. The orifice was changed to a round hole, No. 75 drill, instead of a slot. The pressure in the blade holder was changed from atmospheric to 500 lb per sq in. gage, thus giving a pressure drop of 100 lb across the nozzle. The amount of steam passing through the nozzle per hour was the same as in the previous lower-pressure tests.

When these tests were started the blades were found to be coated with a film of what appeared to be magnetic oxide. This coating formed all over the blade and appeared to be the result of a reaction between the superheated steam and the steel in the blade. When the pressure was dropped to 300 lb and the temperature to 500 F, this action stopped. Stainless-steel blades, 18 chromium, 8 nickel, were substituted for the steel blades. When the steam was of high quality and apparently contained no sodium hydroxide, the blade remained clean and was not attacked by the steam. When the boiler water contained sodium hydroxide the blades were attacked and became so brittle that they broke upon being removed. Substitution of monel-metal blades appeared to overcome these difficulties and a series of tests was run with this type of blade. The results of these tests are given in Table 4.

PLANT EXPERIMENTS

The results reported from many power plants indicate that the detection of the difficulty encountered with blade deposits is mainly in the form of loss of capacity. The efficiency loss becomes appreciable before capacity loss is detected. When capacity loss is noticed the deposits have formed, and they must be removed to obtain normal operation. Thus if the water treatment is to be studied or modified to prevent the deposits from forming, it is advisable to have a rapid method of studying the effect on the formation or prevention of deposits. Early in the research, attempts were made to devise a small testing unit which could be used to test samples of the steam in an operating plant and tell in a relatively short period of testing whether the steam would cause blade deposits or not. If such a unit could be devised, a definite study could be made on the effect of modification of water treatment on the deposits formed without waiting for the effect to be noticeable at the turbine.

A small test unit was made which would allow a small amount of the steam from the heater just ahead the turbine to flow through a nozzle, strike a section of regular turbine blades, and then exhaust through a regulating valve. With data relative to the temperature, pressure, and size of nozzle available, the velocity of the steam passing the blade and the quantity per unit of time could be calculated. The blade could be removed at any time desired and the amount of deposit formed determined by weighing.

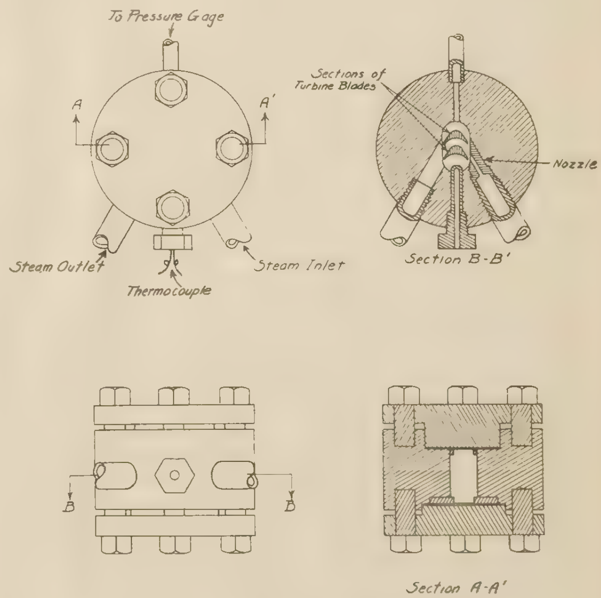


FIG. 6 TEST UNIT USED IN PLANT TESTS

Fig. 6 is a drawing of the test unit developed and used in the plant tests. The steam passed through a nozzle and over sections of regular turbine blades. Leaving the blades, the steam passed out of the unit and was piped through a valve to a desired point for disposal. By regulation of the exhaust valve the pressure in the chamber holding the blade could be held at any desired value. The temperature was determined by means of a thermocouple inserted in a well. The blade was cleaned, dried, weighed, and inserted into the holder. The cover was then bolted on and the steam pressure adjusted to the desired value. At th

TABLE 5 RESULTS OF TURBINE-BLADE DEPOSIT DETECTOR TESTS

		Steam conditions					Blade deposit			Boiler-water analysis					
Test no.	Starting date	Flow, lb per hr	Nozzle Press, lb per sq in.	Nozzle Temp, F	Total flow, lb (thousands)	Duration of test, hr	No. 1 blade g	No. 2 blade g	Mg per million lb steam	Dissolved solids	Total alkali as Na_2CO_3	PO_4	pH	Na_2SO_4	Ratio, Na_2SO_4 Total
1	4/11/34	870	346	668	45	53	0.0060	0.0053	251	170	74	52	10.2	26	0.1
2	4/16/34	870	345	666	205	234	0.0244	0.0211	221	182	70	50	10.1	28	0.3
3	4/26/34	430	415	686	112	260	0.0178	0.0175	310	198	83	44	10.2	30	0.4
4	5/ 8/34	430	415	675	22	52	0.0008	0.0013	...	203	86	46	10.1	31	0.4
5*T*	10/25/34	870	415	665	102	117	0.0014	0.0053	66	...	140	80	10.7	640	4.6
6*T*	10/30/34	870	415	660	119	137	0.0008	0.0016	20	...	150	75	10.8	655	4.3
7*T*	11/ 5/34	870	415	670	87	100	0.0006	0.0032	44	...	160	95	10.9	630	3.9
8*T*	11/12/34	870	415	675	65	75	0.0004	0.0009	20	...	180	95	10.9	625	3.5
9*T*	11/15/34	870	415	670	166	191	0.0014	0.0028	31	...	170	60	10.9	500	2.9
10*T*	11/23/34	870	415	660	143	164	0.0007	0.0011	13	...	120	60	10.8	470	3.9
11*T*	11/30/34	870 and 460	415	650	89	167	0.0156	0.0060	242	...	170	65	10.9	580	3.4
12*T*	12/ 7/34	460	415	640	75	162	0.0008	0.0011	25	...	190	60	11.0	420	2.3
13*T*	12/17/34	460	415	635	123	268	0.0008	0.0018	21	...	230	60	11.2	380	1.6
14*T*	12/28/34	460	415	640[1]	154	336	0.0045	0.0081	82	...	250	60	11.1	420	1.6
15*T*	1/11/35	460	415	640[1]	75	163	0.0003	0.0006	12	...	225	50	11.0	475	2.1
16*T*	1/18/35	550	415	640[1]	94	171	0.0017	0.0025	45	...	240	40	11.2	540	2.2
17*T*	1/25/35	550	415	640[1]	93	168	0.0064	0.0097	173	...	230	40	11.1	590	2.5
18*T*	2/ 4/35	550	415	640[1]	54	99	0.0002	0.0002	7	...	195	45	11.1	500	2.5
19*T*	2/ 8/35	460	415	640[1]	77	167	0.0014	0.0024	49	...	190	40	11.1	540	2.8
20*T*	2/15/35	460	415	640[1]	65	141	0.0010	0.0036	71	...	195	40	11.1	530	2.7
21*T*	2/21/35	460	415	640[1]	88	192	0.0044	0.0058	116	...	230	60	11.1	510	2.2

[1] Approximate.

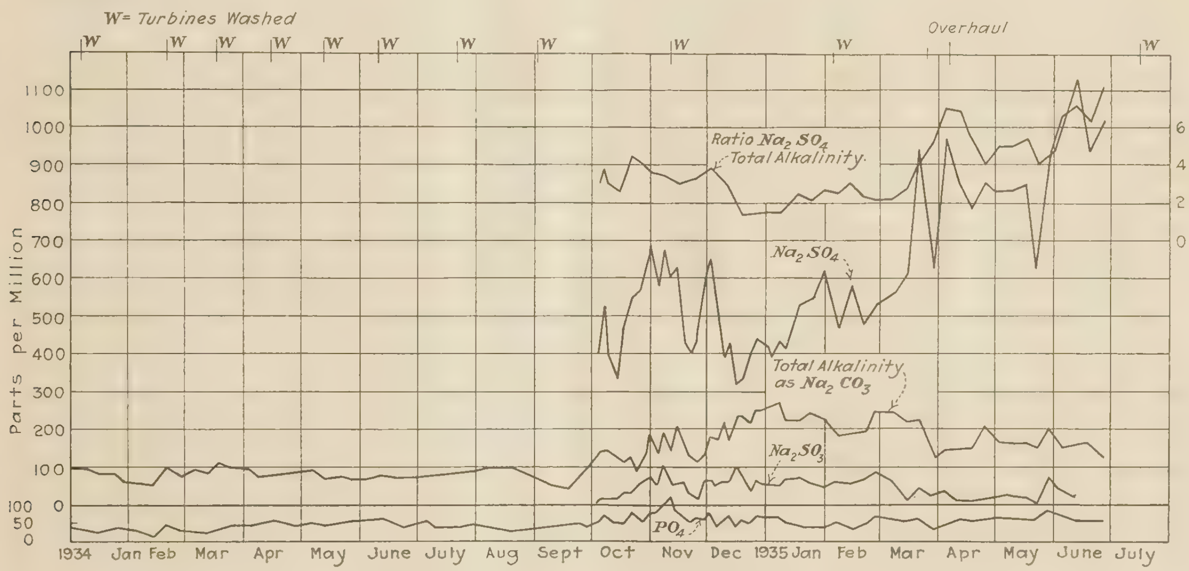


FIG. 7 AVERAGE ANALYSES OF ALL BOILERS IN SERVICE

end of the test the steam was shut off, the blade removed while still hot, cooled in a desiccator, and weighed. The amount of steam passing through the unit was calculated and the amount of deposit per unit of steam was determined. The deposit formed was reported in milligrams per million pounds of steam passing through the unit.

The unit was tested in a central plant operating at a pressure of 600 lb per sq in. and a temperature of 725 F. The steam pressure at the nozzle was between 350 and 520 lb per sq in. and the temperature between 635 and 725 F. The pressure drop through the unit was adjusted so as to obtain the desired rate of flow. This was held at 870 lb per hr in the majority of tests and was accomplished with a pressure drop of 100 lb. With this pressure drop the velocity of the steam passing the blades was about 460 fps. Table 5 gives the results of the tests run with the test unit.

The unit was installed on a line connected to the steam header just ahead of the turbine. The first test was run for 53 hr and gave a total deposit of 11.3 mg or at the rate of 251 mg per million pounds of steam. The next test was run under the same conditions for 234 hr and the amount of deposit was 45.5 mg and the rate of deposit was 221 mg per million pounds of steam. This indicated that the rate was almost constant and a test of 52 hr was sufficient to detect the amount of deposit forming.

In the next test the rate of flow was lowered from 870 to 430 lb per hr, which changed the velocity from 460 to 140 fps. The deposit increased forming at a rate of 310 mg per million pounds of steam. This third test showed that the deposit would form over a wide range of steam flow through the unit. During these tests the regular water treatment was in use and the plant was experiencing difficulty with blade deposits. The turbine was washed every four weeks.

Fig. 7 shows the average boiler-water conditions during these tests and the time of turbine washings. The boiler-water treatment was modified early in October, 1934, to increase the sulphate concentrations so that they would be similar to those found advisable in the laboratory tests. In order to obtain the desired sulphate, sodium sulphite was added. The sulphite would react with the small amount of oxygen present and reduce corrosion. The sulphate formed would aid in the prevention of blade deposits.

The sodium-sulphate content before the addition of the sul-

phite was between 15 and 30 ppm and is not shown. At the time of change in water treatment the sodium-hydroxide content was also increased with a corresponding change in pH value. All of these changes increased the total dissolved solids in the boiler water from a maximum of 260 ppm to about 1200 ppm. The ratio of the sodium sulphate to total alkalinity increased to a range between 3.5 and 4.6 for the period between October 1 and November 11.

The turbine was washed on July 22 and not washed until Sept. 2, a duration of six weeks instead of four weeks as usual. The loss due to blade deposits, calculated from increase in stage pressure over that of clean turbine, was greater at the end of these six weeks than for a regular four-week period, as was to be expected. The water treatment was changed about October 1, a period of four weeks after the turbine had been washed. At this time the tests of the turbine showed about the regular amount of loss due to deposits. The turbine was not washed. The deposit did not appear to increase as rapidly as before the changes in water treatment and the turbine was allowed to run until November 11 before washing. The deposit did not appear to be causing any more difficulty at the end of the ten weeks than at the end of the first four weeks.

Tests run on the small testing unit during the period after change in water treatment showed a deposit rate of between 20 and 66 mg per million pounds of steam as compared to 250 mg per million pounds of steam prior to the change, a reduction of about 90 per cent in the deposit. The amounts of deposits weighed were about 0.5 mg which was about the limit of accuracy of weighing. These results appeared to indicate that the deposit was being materially reduced.

The turbine was not washed again until February 3, a duration of twelve weeks. During this period the sodium sulphate to total-alkalinity ratio dropped from 3.5 to 1.6 and was 2.5 at the time of washing. The turbine was taken out of service on March 24, for overhauling and inspection, and was washed to aid in cooling. When the turbine was opened it was found free from deposits, indicating that if any deposits were forming they were all removed by washing.

While the turbine was out of service the method of adding the sodium sulphite was changed from intermittent dosage to a continuous feed. It was hoped that this would give a better control of chemical feed and thus show a more consistent sulphate concentration. However, the sulphite was added at a point where appreciable oxygen was present, ahead of the heaters, and this brought about an increase in the sulphate formed. The sulphate content increased to a much higher value than was previously maintained.

When the turbine went back into service blade-deposit formation appeared to be slow. The turbine was not washed again until July 14, a duration of 14 weeks. Thus the time of washing was changed from four-week periods prior to the change in water treatment to an average of 12 weeks; and this change resulted during the period when the control of the water treatment was in the experimental stage.

DISCUSSION OF RESULTS

The results of these studies indicate that the turbine-blade fouling is caused by chemicals carried into the steam from the boiler water. The amount of this contamination does not have to be large and steam which would normally be referred to as being of excellent quality might cause a great amount of deposit. The chemical causing the major portion of the difficulty appears to be sodium hydroxide. This is present in practically all boiler waters and when carried with steam that is superheated forms a concentrated sticky solution. This sticky material forms the basic binder and adheres to the turbine blades. Other salts present in the form of a dry powder would normally be swept past the blades. However, when caustic soda is present these salts adhere with the pasty caustic. This may cause a deposit in which the relative proportions of the constituents bear no relationship to the ratios in which they occur in the boiler water.

The amount of deposit is not proportional to the carry over from the boiler, but depends upon the relationship which exists between the various salts in the boiler water. Thus high hydroxide in respect to the total solids may cause an excessive deposition even with a low concentration of sodium hydroxide in the boiler water. However, a boiler water with a sodium-hydroxide content low with respect to the total solids present but high in concentration might not cause a deposit. This is because salts other than sodium hydroxide apparently aid in the prevention of the deposits.

The salts which form a dry powder in the superheated steam apparently adhere to the surface of the droplet of concentrated pasty sodium hydroxide, and if present in sufficient amount entirely coat the particle and allow it to pass through the turbine without sticking to the blades. Such a method of prevention naturally requires large amounts of the dry powder in proportion to the sticky material. The plant tests and laboratory experiments indicate that if sodium sulphate is to be used to prevent turbine fouling it must be present in amounts greater than four or five times the actual sodium hydroxide present. Power-plant operators hesitate to increase the total solids in the boiler water because they fear carry over. However, if this is properly controlled and the carry over does not become excessive, the deposits will be prevented.

The application of the inorganic-salt treatment to one large central power plant while still in the experimental stage showed that the period of turbine washings were reduced from four weeks to about twelve weeks, or from twelve times a year to four. It is quite possible that when the water treatment is better understood washing in this plant may be entirely eliminated. The cost of the treatment used has been low, less than $500 per year, and the savings have been materially high since corrosion has also been reduced.

The application of these data to power-plant operation should be made only after a careful study of the conditions existing at the particular plant and changes which are made should be under the control of a chemical engineer who thoroughly understands this field of water treatment.

CONCLUSIONS

A summary of conclusions drawn from the results of this work is as follows:

(1) The basic material causing turbine-blade fouling is sodium hydroxide.

(2) Inorganic salts, such as sulphate, chloride, and carbonates, if present in sufficient amounts, will prevent turbine-blade fouling.

Cavitation Testing of Model Hydraulic Turbines and Its Bearing on Design and Operation

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The author discusses cavitation tests made with model turbines at the Holtwood hydraulic laboratory of the Pennsylvania Water and Power Company and points out benefits which have been realized therefrom. In addition to presenting the results of these tests the author compares them with results obtained by other investigators in this field. The author presents his views regarding the further application of the equipment in the Holtwood laboratory for the purpose of investigating various elements in the design of turbines which may have some bearing on their cavitation characteristics. He points out the need of coordinating the results of cavitation investigations now being conducted in various laboratories so that the industry as a whole may be benefited thereby.

THE Holtwood hydraulic laboratory was put into operation early in the summer of 1930 and since that time has been in almost constant use. Although most of the work done in the laboratory has been in connection with the Safe Harbor development, turbine models of three large projects have been tested for manufacturers on a rental basis.

Since the initial Safe Harbor development is now complete, it seems desirable to review the accomplishments of the laboratory and to point out what benefits have been realized from the research which has been conducted.

The laboratory shown in Fig. 1 is laid out to accommodate normally a model 16 in. in diameter. The pressure chamber is 12 ft by 18 ft inside, so that scroll cases of the desired design may be constructed of wood or other material.

The wheel setting is vertical, and directly below the runner there is an observation chamber in which elbow draft tubes of the shape desired may be constructed. The discharge passes a diaphragm separating the observation chamber from a vacuum chamber, so called because it may be necessary to produce a vacuum in the space above the water in order to maintain the level sufficiently high to seal the draft tube. The set-up allows sufficient flexibility so that spreading tubes, or even straight conical tubes, may be tested, providing a collector is used to direct the discharge into the vacuum chamber. A more detailed description of the laboratory may be found in two previously published papers.[2,3]

In making complete tests of a model the procedure is first to determine the efficiency characteristics in the usual manner, except that with a movable-blade runner there is one more variable to be considered. Tests are made by adjusting the dynamometer to cover the range of speeds desired, observations being taken of the speed, the quantity of water used, the power output, and the head. The water quantity is measured by a 36-in. and 15-in. venturi meter, the output of the wheel by an electrical dynamometer capable of absorbing 300 hp, the speed by a synchronous timer driven by an a-c generator geared directly to the dynamometer shaft, the headwater elevation by a mercury manometer connected to piezometers in the pressure chamber, and the tailwater elevation by a float gage connected to pressure orifices in the vacuum chamber.

For each blade angle, curves are drawn as shown in Fig. 2 for appropriate gate openings, selecting these gate openings so that the most efficient combination of blade and gate will be bracketed. By adjusting the torque absorbed by the dynamometer, the speed may be varied over the desired range which is dictated by the need of including the extreme values of ϕ, which is the ratio between the peripheral speed of the runner and the theoretical spouting velocity of the water, which corresponds to the synchronous speed of the prototype and the maximum and minimum heads at which it may be operated. These tests are run at a very low, or even negative, draft head in order to create as great a pressure around the blading as possible, and thus preclude any likelihood of the results' being affected by separation between the water and the blades, which is cavitation.

After a series of such tests, one for each blade angle, the complete efficiency characteristics of the unit can be plotted. In this way the efficiency curves for the values of ϕ for the various heads in the prototype may be determined. More important, as far as subsequent cavitation tests are concerned, is the selection of the gate openings which will bracket the best combinations of blade and gate for each value of ϕ.

Tests must then be repeated for each value of blade and ϕ with individual series of tests for at least three gate openings; but while the foregoing are held constant, the headwater and tailwater levels are varied to change the ratio between draft head and total head. Any failure to maintain absolutely the right speed to correspond with the head actually used may be corrected on the basis of information obtained from the previous test in the same way that field tests are corrected to guaranteed head by means of the characteristics determined from model tests.

As the draft head becomes greater there will be a tendency for the water to pull away from the underside of the blades with consequent alteration of the efficiency and output characteristics as determined in the first series of tests. Therefore, the object of this second series is to measure to what extent the draft head can be increased without causing this so-called cavitation. It is consequently necessary to adopt some criterion which may be

[1] Hydraulic Test Engineer, Pennsylvania Water and Power Company. Mr. Davis received his C.E. degree at Cornell University in 1923. For five years following graduation he was in the employ of the Niagara Falls Power Company as engineer of hydraulics. The next two years he spent on hydro-power-plant design for the Aluminum Company of America. Since 1930 he has been associated with his present concern, during which period the Safe Harbor turbines were designed.

[2] "Safe Harbor Kaplan Turbines," by G. W. Spaulding and L. M. Davis, *Electrical Engineering*, October, 1932, vol. 51, pp. 728–733.

[3] "Model Testing at Holtwood Hydraulic Laboratory," by L. M. Davis, published by Pennsylvania Water & Power Company, Holtwood, Pa., May, 1934.

Contributed by the Hydraulic Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers to be held in New York, N. Y., December 2 to 6, 1935.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

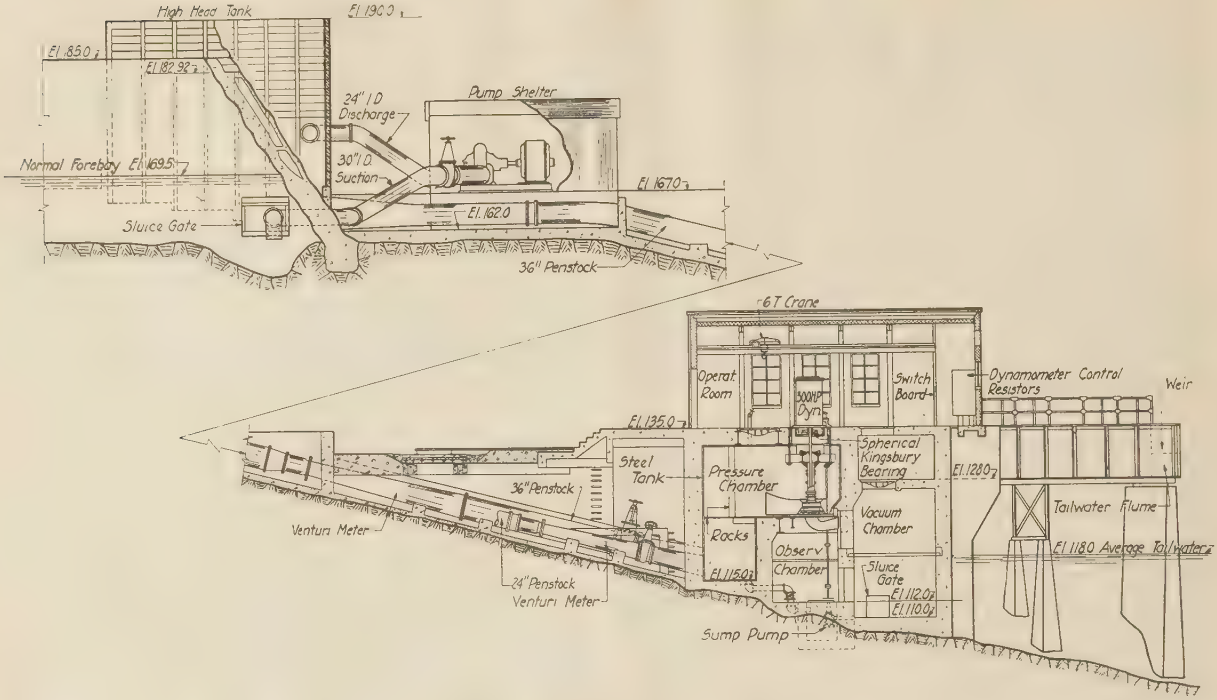

Fig. 1 The Holtwood Hydraulic Laboratory

used for defining the conditions under which a given model will just begin to cavitate.

L. F. Moody and F. H. Rogers have offered[4] a cavitation coefficient K_c which is obtained from the equation

$$K_c = [h_a - h_m - h_s - (E_d C_2^2/2g)]/H$$

where h_a is the atmospheric-pressure head, ft; h_m is the absolute pressure at the point of minimum pressure, ft; h_s is the static draft head, ft; E_d is the draft-tube efficiency; C_2 is the absolute-discharge velocity from the runner, fps; g is the acceleration due to gravity, ft per sec per sec; and H is the total effective head on the turbine, ft.

Inasmuch as cavitation is assumed to take place only when the lowest pressure h_m becomes equal to the vapor pressure, this substitution may be made, and the value to be used is a function of water temperature alone. To obtain a numerical value of K_c it is necessary also to know the draft-tube efficiency and the absolute discharge velocity. It should be noted that since all the terms in the right-hand side of the equation are expressed in feet of water, K_c is dimensionless.

D. Thoma originated the cavitation coefficient σ through his work at Munich. The physical meaning of this coefficient can be visualized by imagining the following sequence of events. Suppose the model turbine to be operating without cavitation under a constant head and with constant discharge. Since the forebay and tailrace elevations are variable, the tailrace can be lowered while the total head is maintained constant. As the draft head is thus increased, the pressure on certain parts of the runner blade will decrease. When the minimum pressure at any point on the blades just reaches the vapor pressure H_v, cavitation is said to begin. Suppose the unit is shut down as soon as this condition is reached. After the turbine gates are closed, the absolute pressure on the blades is obviously the barometric height H_b minus the draft head H_s. In the operation of the unit, the pressure at some point on the blades was, therefore, lowered from

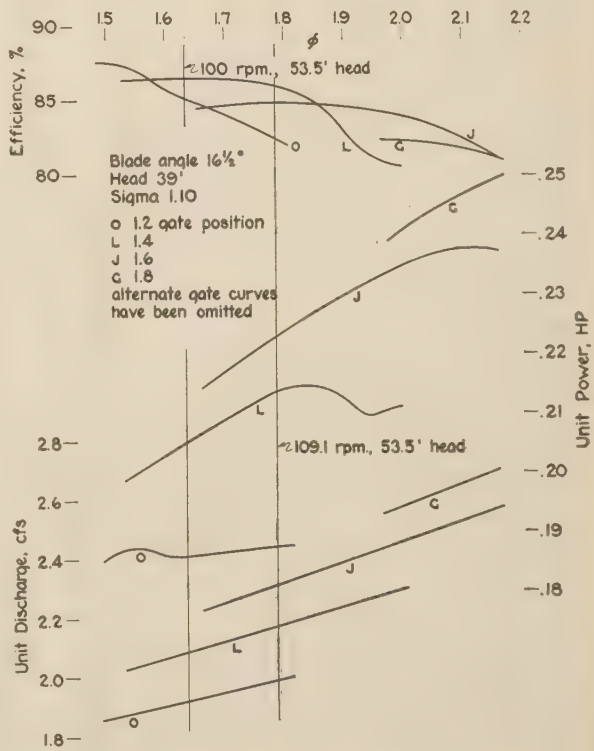

Fig. 2 Typical Results of ϕ-Efficiency Tests

[4] "Inter-Relation of Operation and Design of Hydraulic Turbines," by L. F. Moody and F. H. Rogers, *Engineers and Engineering*, July, 1925, vol. 42, no. 7, pp. 169–187.

$H_b - H_s$ to the vapor pressure H_v. This decrease in pressure is a function of velocity and in fact is proportional to the square of the discharge of the unit. Since the square of the discharge is proportional to the head, it follows that the reduction in pressure, $H_b - H_s - H_v$, is itself proportional to the head. Therefore, it is possible to use in place of the pressure drop $H_b - H_s - H_v$, which applies only to the conditions of the test, the ratio $(H_b - H_s - H_v)/H_t$, which applies to all heads and which is identified as σ.

Comparison between the Moody and the Thoma formulas shows marked similarity. The h_a of the former is the same as H_b, or barometric pressure, in the latter. Likewise the static draft head h_s is the same as the suction head H_s. It has already been pointed out that h_m is equal to H_v. Consequently, the only difference lies in Professor Moody's term $E_d C_2^2/2g$ which is omitted by Dr. Thoma. The inclusion of this term simply means K_c applies to the runner alone since correction is therein made for the effectiveness of the draft tube, whereas σ applies to the wheel and setting as a whole.

The results of the cavitation or σ tests are shown in Fig. 3, where again efficiency, horsepower, and discharge are plotted, but this time as functions of σ. These curves represent the results of three gate openings corresponding to but a single combination of blade angle and ϕ, and similar tests must be conducted to cover all others in the range desired. Ordinarily, only the higher blade angles are of importance and from three to five values of ϕ are used generally to investigate fully the cavitation characteristics for the entire range of head used in the finished plant. From complete cavitation information at four or five values of ϕ

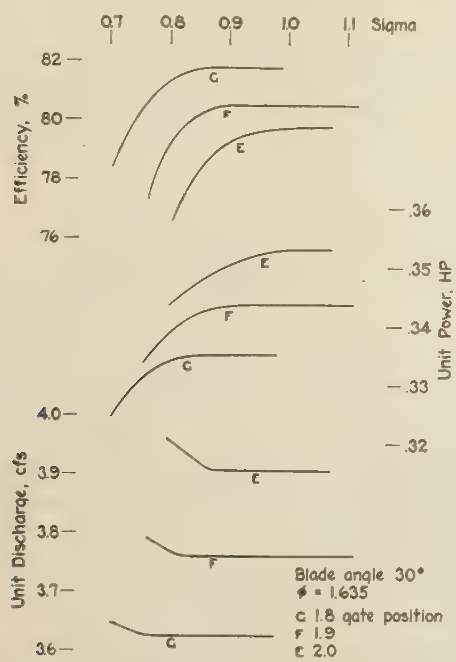

FIG. 3 TYPICAL RESULTS OF σ OR CAVITATION TESTS

it is possible to cross-plot and obtain data for any intermediate value.

It will be noted that on the right of Fig. 3 all of these curves become horizontal. The constant values correspond to those which have been observed in the preliminary ϕ-efficiency tests.

As σ becomes smaller with greater draft head, there occurs a change in the ordinates. The point at which this occurs is known as the break in the curve, or the beginning of cavitation. The exact determination of this point depends somewhat on judgment, and some variation will be found therein, but if several persons pick these points and results are plotted to form smooth curves, independent interpretations of the same data will agree satisfactorily.

One criterion of the break is the dropping off in efficiency. Another is the increase in discharge. These will not coincide precisely on account of the fact observed by Dr. Thoma that there is a slight tendency for efficiency to increase just as cavitation

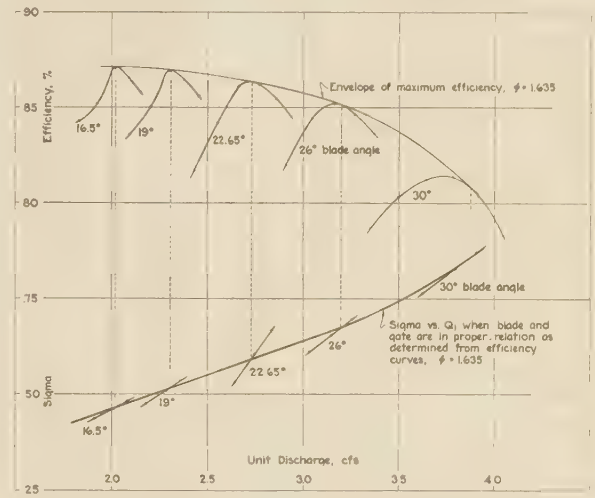

FIG. 4 RECAPITULATION OF σ-TEST RESULTS FOR ϕ = 1.635

starts which, he explains, is probably due to the decrease in skin friction.

However, it has been observed that with some models the efficiency tends to decrease at a higher value of σ than the point where the unit discharge starts to increase. It is felt that this is due to faulty design resulting in local cavitation which takes place at one or more points on the blade at draft heads somewhat lower than the draft head which causes cavitation over a general area.

It has been the practice in the Holtwood hydraulic laboratory to use the break in the unit-discharge curve to determine critical σ. If there is any great discrepancy between the break in the efficiency curve and the break in the unit-discharge curve, the more conservative should be used, although it is felt that this is an indication of defective design, and possible alterations to correct this discrepancy should be investigated.

The question may be asked whether the break in the σ curve is affected by the head at which the test is conducted. Theoretically there should be no effect and tests made at one head should be applicable to others. This matter was investigated by Prof. Wilhelm Spannhake, of Karlsruhe, whose laboratory was adapted particularly for experiments covering a wide range of heads from about 33 ft down to 5 ft. It was found that especially at the very low heads there was a marked effect, but that for higher heads the break in σ reaches practically a constant value which is independent of further changes. Similar tests at much higher heads were performed at Holtwood without revealing any appreciable difference in σ at which the breaks occur. Therefore, it seems preferable, but not essential, that the cavitation test be made at a head closely approximating that of the prototype.

The next step is to recapitulate the results of the foregoing tests. This is done by plotting the value of σ corresponding to the break against unit discharge as shown in Fig. 4. In order to determine the value of σ at which cavitation begins when the unit is operated at the correct relation between gate and blade, the envelope of the individual efficiency curves is drawn, and

vertical lines are dropped from the points of tangency. The intersections of these verticals with the σ versus unit-discharge curves for the corresponding blade angles locate points on the final curve representing the relation between σ and discharge for a Kaplan unit when operating with the blade and gate in the proper relation to each other. This procedure is repeated for each value of ϕ, thus giving in the end the plot shown in Fig. 5. By cross-plotting, curves may be obtained for any intermediate values of ϕ.

The group of curves given in Fig. 5 gives the complete cavita-

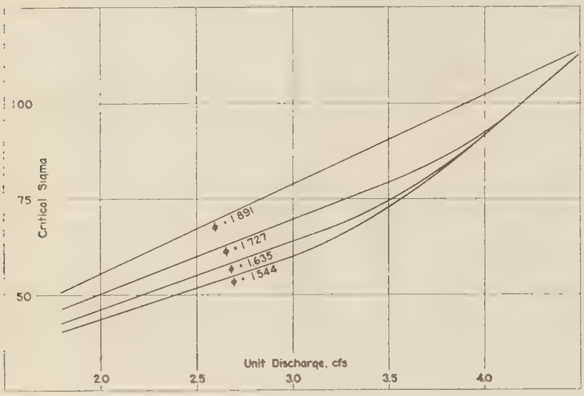

Fig. 5 σ Versus Unit Discharge for Various Values of ϕ

tion characteristics of the model for the range of ϕ indicated. From these curves the proper setting of a homologous runner for any installation with reference to tailwater for a given power output may be determined.

One general characteristic of all tests so far has been the marked effect which the increasing of ϕ has had in aggravating the danger of cavitation. The practical application of knowledge gained through tests is illustrated by the adoption of a lower rotational speed for the most recent installation at Safe Harbor. This reduction proves to have resulted in a substantial increase in capacity in spite of the fact that judging only from the output-ϕ characteristics of the model, the capacity would have been adversely affected. Therefore, it is apparent that when cavitation becomes a limiting factor, the urge for higher speeds for the sake of greater capacity and lower generator cost must be tempered. Since it is obvious that, without cavitation, increasing speed gives greater physical capacity, while the effect of increasing speed on power output as limited by cavitation is just the opposite, it becomes exceedingly important to explore completely the characteristics of a model before making a decision regarding the speed of rotation.

The proper setting for the turbine with reference to tailwater elevation can be determined from Fig. 5 as soon as the runner diameter, speed, rated power, and total head are established. From the efficiency tests, the expected efficiency at the rated head and output may be determined and the discharge computed. The unit discharge may then be obtained, and from Fig. 5 the critical σ will be found. The values of all the terms in the formula for σ have now been obtained except the draft head H_D, which is readily obtained. Adding this value of the draft head to the tailwater elevation gives the runner elevation which will give the desired output without cavitation. Since cavitation would be expected to result if the maximum output as determined by test were slightly exceeded, it is customary, as a factor of safety, to lower arbitrarily the setting a small amount.

Not only is the determination of the critical σ, which is the value of σ at which cavitation begins, useful in fixing the proper height of setting, but conversely the laboratory results can be used to determine the allowable output under given conditions of forebay and tailwater level after the setting has been established.

At this point distinction should be made between critical σ as applied to the runner and plant σ which refers to the plant. Both are expressed by the same formula, but obviously plant σ must not be less than critical σ if avoidance of cavitation is to be expected. It is also clear that by assuming the critical value of σ equal to the plant σ for any given H_s and H_t, the maximum allowable unit discharge can be read directly from the scale of abscissas in Fig. 5.

Whereas it is probably correct, since cavitation is logically a function of water quantity, to derive the discharge for the prototype directly by stepping up from the model without allowance for improved efficiency, nevertheless hydraulic engineers are faced with a serious problem in determining the corresponding limiting power. In the investigations which have been made on the subject of the relation between maximum efficiency of the model and the prototype, no information has been made available

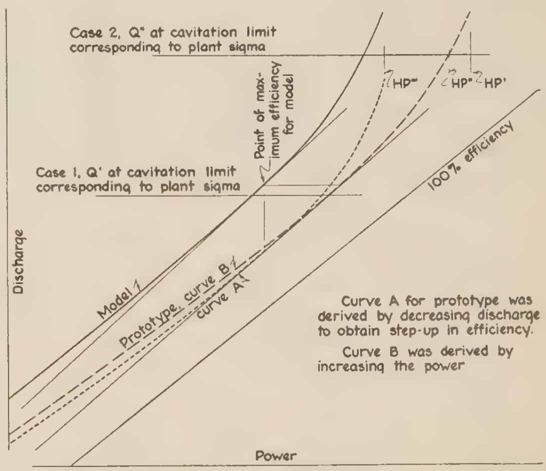

Fig. 6 Effect of Step-Up on Interpretation of Model-Test Results

as to whether the increased efficiency is obtained as a result of increased output from the same quantity of water, or decreased quantity for the same output. Perhaps there may be a combination of the two effects. It will be seen from Fig. 6, however, that as a practical matter it ordinarily makes very little difference whether better efficiency results from lowering the power-discharge curve or by shifting it to the right except for values near the extreme end of the curve. Thus, to find the prototype output corresponding to a discharge of Q', case 1, it makes no appreciable difference whether we use curve A derived by reduction in discharge, or curve B derived from increase in output. However, for a discharge of Q'', case 2, there may be some doubt as to whether cavitation will reduce the output from HP' to HP'' or whether HP'' would still be above the physical capacity HP''', and, therefore, not a limiting consideration.

Therefore, it is sufficient in determining the output as limited by cavitation to assume increase in output in proportion to the expected rise in efficiency resulting from the computed reduction in hydraulic losses. On the other hand, there is great need for further investigation as to the exact nature of this shift by comparison of the discharge versus output curves for various units which have been tested in the field. It is no longer sufficient for our purpose to use a formula by which an increase in maximum efficiency alone can be estimated. There is obviously a need for

further investigation before it may be considered that the principles of stepping-up performance curves are well understood. Uncertainty in this respect is apparent from the fact that the application of the formulas of three separate authorities to the result of the test on one model which has come to our attention is reported to have given answers differing by as much as 4 per cent.

A convenient form for presenting the limiting power output for various head conditions is to draw constant limiting power-output lines using tailrace elevations as ordinates and forebay elevations as abscissas. To obtain this curve it is well first to develop a curve of limiting power output versus gross head, drawing lines of constant forebay elevation. Such a chart is useful in plant operation so that with varying conditions the risk involved in approaching the critical σ of the runner is always the same.

As cavitation data become available it is desirable to compare them so that progress in design may be noted and conclusions drawn as to the features which are or are not advantageous. Fig. 7 shows a plot of critical σ versus unit discharge at rated ϕ for several models which have been tested in the Holtwood laboratory, together with other tests which have come to the author's attention.

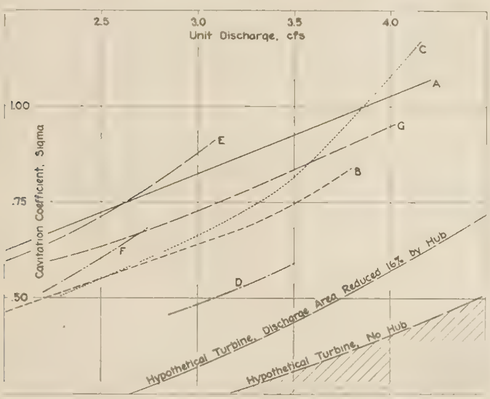

Fig. 7 Comparison of Cavitation Tests on Several Models on the Basis of Unit Discharge

While improvement can be expected with experience, this progress, like efficiency, cannot go on indefinitely. There must be some limit beyond which further improvement is impossible.

To illustrate this principle, there may be imagined a hypothetical turbine in which there would be no drop in pressure below the static pressure as the result of pressure difference on the two sides of the blade or irregularities of the pressure distribution along the blade; although this is clearly impossible, it is at least a limiting case. In spite of this, there would be a reduction in pressure simply by reason of the velocity through the throat. In other words the $(H_b - H_v - H_s)$, previously mentioned as the reduction of pressure caused by the operation of the turbine, now becomes only the velocity head of the water passing through the throat. Assuming that this throat may be perfect as an orifice, that the velocity is axial and equally distributed, and also that there is no reduction in area on account of the runner hub, it could then be shown that when H_t is equal to unity this drop in pressure would be equal to the velocity head $c^2/2g$.

Thus, since $Q_1 = ac$ and $a = \pi/4$

$$\sigma = (4Q_1)^2 / 2g\pi^2$$

The values of σ corresponding to various values of Q_1 are shown by the lowest curve in Fig. 7 which illustrates the ultimate limit of perfection. Should we assume, by way of example, a 16 per cent reduction in discharge area by reason of the runner hub, this limit obviously would be raised still further. Keeping this in mind, progress toward the ultimate goal may be judged. The economic importance of any move in this direction will be seen from a study of Fig. 7 where, at a value of 0.75 for plant σ, the gain in allowable power output for curve B over curve A is approximately 27 per cent. In a new installation for 35-ft rated head and a unit discharge of 2.6 cfs the turbine could be set 6 ft higher for curve B than for curve A.

F. H. Rogers and R. E. B. Sharp, in a paper[5] presented at the Cincinnati meeting of the A.S.M.E., June, 1935, discussed a relationship between plant σ and specific speed derived from a comparison of nine important propeller installations in America and Europe, as shown in their curve Fig. 7,[5] which is reproduced in Fig. 8 of this paper for comparison with the results of the models tested at Holtwood and other laboratories. In the discussion of the paper[5] by Rogers and Sharp, R. V. Terry, hydraulic engineer of the Newport News Shipbuilding & Dry Dock Company, suggested the relation $\sigma = N_s^2/25{,}600$ as a close approximation to the critical σ for normal design of turbines. For small installations where cavitation tests are not warranted, he suggests using a more conservative relation, $\sigma = N_s^2/22{,}500$, for determining the turbine setting with reference to tailwater elevations. It appears from test D, Fig. 8, that actually there are potentialities which may be realized as a result of further development and research and that a substantial improvement in design can be made over that which might be expected from the three relationships.

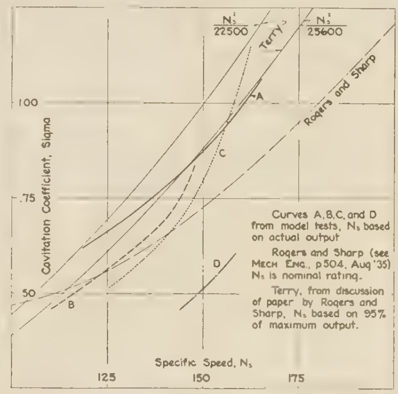

Fig. 8 Comparison of Cavitation Tests on the Basis of Specific Speed

Benefits Derived From Tests

It would be well now to consider the use that has been made of the laboratory results and the benefits derived from the tests made in cooperation with the manufacturers in connection with the design and improvements on the Safe Harbor units.

As a preliminary step in cavitation testing, there is the derivation of complete ϕ-efficiency characteristics of the model. In important installations, strict accounting for the use of water is necessary for reasons of record, and for economy in the preparation of operating schedules. It is seldom possible, without unwarranted expense and loss of time in waiting for extraordinary circumstances, to conduct formal efficiency tests over a sufficient range of heads to gain complete knowledge of the operat-

[5] "45,000-Hp Propeller Turbine for Wheeler Dam," by F. H. Rogers and R. E. B. Sharp, *Mechanical Engineering*, August, 1935, vol. 57, no. 8, pp. 499–506.

ing characteristics of a unit. Consequently, model tests are relied upon in interpreting acceptance-test data to cover the desired variety of conditions to draw up those charts or tables which are necessary in daily operation and in planning the economic loading of units. So much has been written on this subject by other authors that the use of model tests for this purpose seems to be well understood.

An interesting application of model tests in conjunction with index tests in the field is described in a paper by E. T. Schuleen.[6] The reason for conducting the test described by Mr. Schuleen was that in the past there has been considerable uncertainty regarding the effect of runner size on the value of ϕ at which the peak efficiency occurs. Some hydraulic engineers contended that

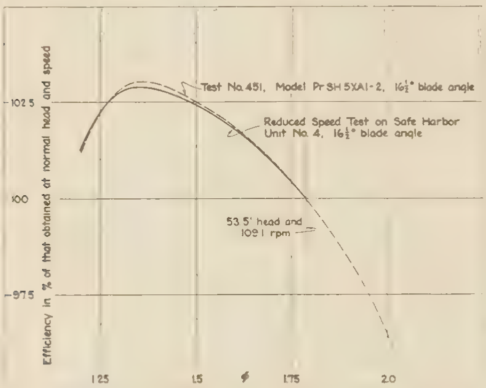

FIG. 9 THE ϕ-EFFICIENCY CHARACTERISTICS OF MODEL AND PROTOTYPE

the peak efficiency occurred at higher values of ϕ as the runner diameter was increased. This question became of vital importance in connection with establishing the speed of Safe Harbor unit No. 2 for 25-cycle railroad service since the model tests indicated an appreciable gain in efficiency at 100 rpm instead of 109.1 rpm, which is the speed of the 60-cycle units. It was consequently decided to test one of the Safe Harbor units at variable speed by means of an artificial load consisting of two other units operated as motors so that a direct comparison could be made between the model and prototype.

Fig. 9 shows a comparison between the envelope efficiencies obtained on this test and the envelope efficiencies obtained on a 16-in. model at the same blade angle of $16^1/_2$ deg. It is felt that this unique experiment proves that the peak efficiency of a given runner design occurs at the same value of ϕ regardless of runner diameter.

Operation of Kaplan units at Safe Harbor, the capacity of which is greater than other units installed so far in this country, has brought up many questions regarding the behavior of these units as affected by cavitation. In seeking an explanation for observed facts and in establishing principles and policies in operation, the laboratory has served as a guide in many cases. This is particularly true in understanding the need for correct setting of the cam for controlling the gates and blades. The importance of correct adjustment and the seriousness of departure from this relation have been brought out clearly by laboratory tests. It is evident from these that whereas error in adjustment on one side of the best blade position may be accompanied by rapid loss in efficiency, on the other side the drop in efficiency is more gradual, but the danger from cavitation is greatly increased.

[6] "Tests at Variable Speeds on a Safe Harbor Turbine," by E. T. Schuleen, published by Pennsylvania Water & Power Company, Holtwood, Pa., May, 1934.

Extensive tests have been made for determining what improvements in blade shape could be made in anticipation of installations of additional capacity. These tests were naturally confined to a narrow range of alterations, being restricted to some blades which were already cast with enough additional finish to allow fairly extensive revamping in machining, and on other blades which could only be chipped to the desired contour. The tests, however, have been sufficiently extensive in their scope to demonstrate the correctness of certain design principles.

Practically every hydraulic engineer who visits the Holtwood laboratory asks: "How do laboratory cavitation tests check actual cavitation conditions at Safe Harbor?" Fig. 10 shows this comparison. The solid line gives the results of the field efficiency tests in per cent of maximum efficiency. The dashed line shows the expected curve from laboratory tests which, as explained previously, are not affected by cavitation. It will be noted that the actual efficiency curve starts to depart from the expected curve at about 33,500 hp. From laboratory tests using the break in the unit-discharge versus σ curve as the criterion for the starting point of cavitation, a limiting output of 34,950 hp was obtained, thus showing a very reasonable agreement.

It should be borne in mind that the test from which Fig. 10 was drawn was conducted when the tailrace was particularly low in order to determine what effect there would be from the worst condition of cavitation. This does not represent ordinary operat-

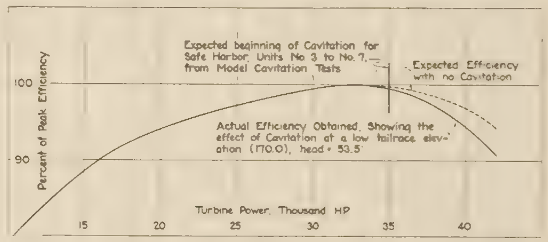

FIG. 10 BREAK IN EFFICIENCY OF PROTOTYPE COMPARED WITH THAT PREDICTED FROM MODEL TESTS

ing conditions and, therefore, the reduction in capacity is not indicative of failure to derive rated output continously from these units. With a normal forebay, the limitation apparently imposed by cavitation in Fig. 10 would be sufficiently raised so that the unit would deliver its full rated output.

It has been found from experience that it is possible to operate somewhat above the limit as determined by the laboratory without accruing enough damage from pitting to be considered out of proportion to the benefits derived from increased output. Inconvenience caused by vibration at these high ratings is successfully combated by the use of a moderate amount of air injected under the head cover.

During the three and one-half years that the Safe Harbor plant has been in operation a certain amount of pitting has occurred. It has been concluded after careful study of the pitting at the time of the numerous turbine inspections and also when the pitted areas were repaired by welding, that the major portion of this pitting resulted from local cavitation induced by irregularities in the blade surface. It is evident that water traveling at high velocities along the blade surfaces cannot follow sudden changes in curvature. When a hollow is present the water leaps over it, forming an unstable cavity. In making repairs great care was taken to grind the welded areas down to a smooth surface free from irregularities or reversals in curvature. Some of these welded portions have been in service two years and no pitting has occurred. This is partly due to the greater resistance to cavitation of the 18 per cent chrome and 8 per cent nickel stainless-steel welding rod which was used, but the improved surface, no

doubt, eliminated local cavitation on the areas which have been repaired and is largely responsible for the absence of pitting.

Further Application of the Holtwood Laboratory Tests

While the accomplishments in the past have proved the wisdom of establishing this laboratory for the purpose for which it was intended, consideration should be given to the future and to what possibilities lie ahead now that the laboratory equipment is available for further work. Up to the present time the experiments have been confined to the change of comparatively few variables, but there still remain for investigation many other elements in the design of a turbine which may have some bearing on its cavitation characteristics, such as the shape of guide vanes, arrangement of the scroll case, form of the head cover, design of the curb ring, influence of the hub diameter, and the runner clearance. Furthermore, there will eventually arise important problems in connection with redevelopment work necessary to secure the maximum amount of additional power from new wheels placed in existing equipment.

The desirability of undertaking further experimentation depends naturally upon what possibilities of improvement are to be expected. Such experimentation is important because it requires but a very slight betterment of either efficiency or cavitation characteristics of the large turbines being built today to justify considerable effort on the part of the designer. A few years ago it was the common impression that because of the high efficiencies attained by Francis turbines, the ultimate goal had been reached. However, with the introduction of the movable-blade propeller, a new field of possibilities was created because it became possible to widen the band of efficient loading. In a like manner cavitation has presented still another field of endeavor wherein the primary object is to make the σ-unit discharge curves, as shown in Fig. 7, approach nearer and nearer to the theoretical limit. It is clearly demonstrated by the admirable performance indicated by curve D, Fig. 7, that actually there are possibilities for a marked advance in this direction.

There is clearly needed some means of coordinating results of different laboratories so that progress may be measured. W. M. White, chief engineer of the Hydraulics Division of the Allis-Chalmers Manufacturing Company, has pointed out[7] his objection to the use of σ. He claims that there is too great a disparity between σ for a propeller turbine such as the Wheeler unit[5] and for a Francis turbine such as used at Boulder dam.[7] In place of σ he proposes the coefficient λ which is simply the expression for the Moody coefficient K_c with the term H omitted. As pointed out previously all of the terms in the formula for the Moody coefficient K_c are expressed in feet of water and therefore, by omitting the divisor H, the result is an expression which indicates simply the pressure available to hold the water to the low-pressure side of the blade. Although Dr. White secures in this way a figure which is more uniform in numerical value, the question may be raised whether or not extreme differences in σ do not express after all the essential difference between the very dissimilar conditions of operation.

Suggestions to relate σ to specific speed have been offered by Messrs. Rogers and Sharp[5] and also R. V. Terry, as mentioned previously. Their criteria have been shown already in comparison with actual tests in Fig. 8. By comparing Fig. 7 with Fig. 8 it is evident that there is a decided shifting of one curve with respect to another resulting from the element of rotational speed which enters into the determination of specific speed.

The ideal comparison would be on the basis of unit horsepower, but in this respect the engineer is still faced with the uncertainty in regard to the applicability of the set-up formula to values other than the point of peak efficiency. Until this problem is solved, the unit-discharge curve is probably the best basis of comparison and should be satisfactory at least for present needs.

A further suggestion comes from the fact that the σ curve is a parabola for the hypothetical turbine, with and without a hub as shown in Fig. 7. It seems logical then to attempt to plot σ against the square of the unit discharge as shown in Fig. 11. If this is done, the hypothetical curves obviously become straight lines.

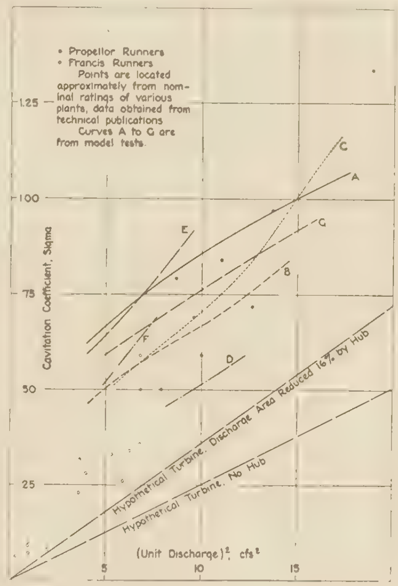

Fig. 11 Comparison of Cavitation Tests on the Basis of the Square of Unit Discharge With Addition of Plant σ's for a Number of Existing Installations

Many of the test curves are distinctly convex downward, particularly those which can be considered as the more reliable. Therefore, it would appear in general that if the curves were extended sufficiently in each case there would be an approach to a straight line passing through the origin followed by subsequent departure. Is it not possible that the point of tangency of a straight line through the origin with the curve in question indicates that value of unit discharge to which the blading is best adapted? Is it not possible also that as more data are collected and plotted in this way, considerable light may be thrown on the perplexing problem of correct shaping of the blades?

As further evidence of the hypothesis that σ is a function of the square of the discharge, there has been added to Fig. 11 points representing the approximate plant σ's and unit discharges for important installations of both Francis-type and propeller-type units. These points have been estimated as accurately as possible from data published in engineering literature, but without going back to original sources. Although this is but a rough comparison it is, nevertheless, striking that, over a wide range of conditions, the setting has been selected by experience (for as far as known no cavitation tests have been made), since for Francis runners there seems to be a distinct straight-line relationship between σ and the square of the unit discharge. Furthermore, if approach to the value of σ for the hypothetical turbine is the thing that is desired, the plant σ's for the propeller installations, most of which have been selected from laboratory tests, do not appear to be as advantageous from the point of excavation as those for the installations of Francis units. This raises the question whether or not in the past hydraulic engineers have been inclined to take risks

[7] "The Construction of the 115,000-Hp Boulder Dam Turbines," by W. M. White, *Mechanical Engineering*, September, 1935, vol. 57, no. 9, pp. 539–546.

that are too great in so far as the settings of the Francis wheels are concerned, or whether, on the other hand, there is some inherent reason why propellers would require more conservative values of σ. One element may be the loss of area taken up by the hub. On the other hand, to decrease the diameter of the hub involves, in the case of Kaplan turbines, mechanical problems and difficulties from clearance spaces, while for all propeller turbines the design of the blading near the hub is particularly difficult and can be mastered only by laboratory tests.

The apparent straight-line relation in Fig. 11 indicates that all that is necessary to reconcile the extremely low σ for the unit at Boulder dam with the high plant σ at the Wheeler dam is to divide each by the square of the unit discharge. No attempt is made here to estimate a normal value for this quotient, since the data for the points shown are but rough approximations. Furthermore, only the plants for which figures were most readily available are included, and no attempt has been made to differentiate between plants encountering serious pitting and those at which operation has been entirely satisfactory.

Inasmuch as the σ versus the unit-discharge curve does offer a reasonably satisfactory basis for comparison, it would seem desirable to have the results of the several laboratories in Europe and America coordinated in this manner. The accumulated data would serve a most useful purpose by informing each laboratory whether the results obtained in any one test were extraordinary in any respect, and thus aid in selecting hidden information that may be found from these tests. For example, curve *D* in Fig. 11 taken alone is simply a test result, but in comparison with the entire group of curves in Fig. 11 it is an achievement —an advance in hydraulic design.

Such a compilation might be undertaken by some central agency, such as the National Hydraulic Laboratory, although consideration should be given to whether or not laboratory errors might not render it worthless. Strange to say, the results are affected very little by laboratory errors because the measure of the break in the σ curve is a change of condition. Consequently, errors in power measurement may have any magnitude as long as they are consistent, while errors in discharge measurement would only shift the final curve slightly to the right or left. In this respect the situation differs essentially from the usual model efficiency tests since the results are not sensitive to ordinary errors.

Finally, it is the author's hope that efforts being made in the investigation of cavitation in various laboratories can be coordinated so that the industry as a whole may be benefited. The Holtwood laboratory offers its support to any movement of this kind.

Radiation From Nonluminous Flames

By H. C. HOTTEL,[1] CAMBRIDGE, MASS., and V. C. SMITH,[2] DETROIT, MICH.

Experimental results on the total radiation due to carbon dioxide, water vapor, and mixtures of the two that were recently reported were obtained in a chromel-wound furnace, the maximum operating temperature of which was 1850 F. The final plots were obtained by extrapolating to 3600 F the experimental data covering the range from room temperature to the furnace maximum.

A method is now described for determining these properties directly in the upper temperature range. It consists of measuring the emission of radiation, and the corresponding temperature, from overventilated flames of carbon monoxide, hydrogen, and mixtures of these in a Meker burner of varying length. The products in the flame are calculated from the mixture fed to the burner.

Results are presented for flames of carbon monoxide in mixtures of oxygen and nitrogen in which the burner grid length is varied from 2 to 16 in., the theoretical oxygen from 100 to 300 per cent, and the flame temperature from 2600 to 3800 F. The results establish the validity of the carbon-dioxide radiation chart in its high-temperature range. Less extensive measurements on illuminating-gas flames establish the validity of the water-vapor radiation chart at temperatures of 2500 to 2800 F when the thickness of gas layer is small.

The problem of evaluating the expected radiant-heat transmission from a nonluminous flame has an obvious practical importance in the design of certain types of industrial furnaces, the evaluation of heat transmission in the internal-combustion engine, and studies in the mechanism of combustion of gases. The literature on the subject includes diverse opinions as to the nature of the phenomenon and does not contain data adequate for the substantiation of any proposed method of predicting the magnitude of the flame radiation. Accordingly the present work was undertaken to determine whether one of the proposed theories could be substantiated by experiment.

The radiation emitted from a nonluminous flame is due in part to radiation emitted consequent upon chemical changes of the molecules in the flame, called chemiluminescence; and in part to thermal radiation from the hot products of combustion, such as carbon dioxide and water vapor. The first kind of radiation depends upon the nature of the chemical reactions involved and is approximately proportional to the amount of fuel burned; the second depends on the temperature, size, and shape of the flame and its hot combustion products and on the concentration of radiating constituents in it, and is influenced by the amount of fuel burned only to the extent that the latter affects the temperature distribution in the flame.

The work of Garner and Johnson (1)[3] and of Johnson (2) has indicated that chemiluminescence, though of extreme importance in studying the mechanism of combustion reactions, accounts for a very small portion of the total radiation emitted from a flame. The calculation of total radiation from a nonluminous flame in an industrial furnace depends, consequently, on quantitative knowledge of the magnitude of thermal radiation from those hot products of combustion which are capable of radiating, namely, carbon dioxide and water vapor.

The earliest quantitative measurements of radiation from carbon dioxide and water vapor were made in 1894 by Paschen (3), who mapped the infrared emission and absorption spectra of hot carbon dioxide and water vapor and showed the interrelation between emission and absorption for those gases. Further work by him on the emission spectra of nonluminous flames led him to conclude that the radiation from them was purely thermal in origin and due solely to the carbon dioxide and water vapor in the flames. The work of H. Schmidt (4) in 1909 confirmed the validity of the assumption that radiation from the products of combustion in flames was thermal in character by proving that Kirchoff's law of the equivalence of absorptivity and emissivity (applicable only to thermal radiation), was applicable to radiations from an illuminating-gas flame. These fruitful suggestions of the physicists remained unused by engineers until Schack (5) in 1924 pointed out how data on the infrared absorption spectra of gases could be used to predict the magnitude of emission of radiation from them.

Moeller and Schmick (6) in the following year measured the radiation from small Bunsen flames, making the measurements in such a way as to permit comparison with the calculations of Schack. Although their experimental results were in agreement with the calculations within 5 to 25 per cent, the agreement was not conclusive since (*a*) the flame diameter was so small as to make edge effects large, (*b*) the effect of water vapor in the combustion products was hidden by the much greater effect of carbon dioxide, (*c*) the basis of comparison with theory (i.e., Schack's calculations based on infrared absorption data) was not established with adequate precision.

Recent experimental data have been presented by Hottel and Mangelsdorf (7), giving the total thermal radiation from hot carbon dioxide, water vapor, and mixtures of the two over a wide range of variation of concentrations, and at temperatures up to about 1900 F. By appropriate plotting, the data have been extrapolated to higher temperatures. Precise measurements of the right kind on nonluminous laboratory flames should serve as a basis for testing the numerical accuracy of the carbon-dioxide and water-vapor radiation charts at high temperatures; those charts could then be used for calculations of radiant-heat transmission from large industrial flames.

Measurements of total radiation of the type of Callendar (8), von Helmholtz (9), and Haslam, Lovell, and Hunneman (10) were made with the whole flame in view of the radiation receiving instrument. These are difficult to compare with the calculation of the expected radiation from hot carbon dioxide and water vapor, because knowledge of the variation of temperature and composition in the flames is not available, and because in some

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

[3] Numbers in parentheses refer to similarly numbered items in the bibliography at the end of the paper.

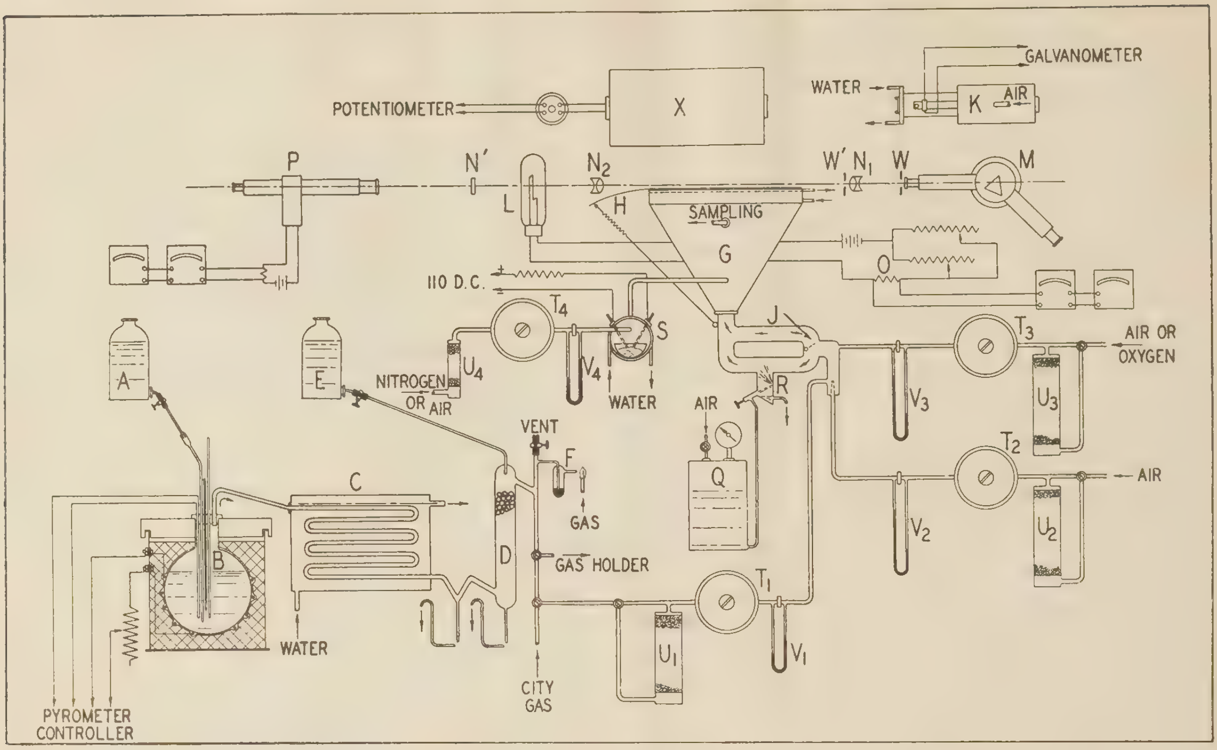


Fig. 1 Apparatus Assembly

A = formic acid	G = burner	N = achromatic lenses	T = pressure regulators
B = carbon-monoxide generator	H = slide	O = standard resistance	U = calcium-chloride chambers
C = condenser	J = butterfly valve	P = optical pyrometer	V = orifice manometers
D = scrubbing tower	K = thermopile	Q = 25 per cent brine	W = diaphragms
E = caustic soda	L = tungsten band lamp	R = spray chamber	X = black-body furnace
F = safety trap	M = spectroscope	S = arc chamber	

cases a considerable portion of the thermal radiation was absorbed by a column of cold gas containing combustion products and occupying the space between the flame and the radiometer. Fishenden and Saunders (11), making a rough calculation of what to expect for one set of conditions studied by Callendar, found agreement with predictions based on thermal radiation inconclusive.

Experimental Procedure

To permit a comparison of experimental results with calculations involving the temperature, composition, and depth of flame it is necessary to burn the gas in a burner of such type as will produce a flame uniform in composition and temperature along a line of sight normal to the direction of flow of the gases; to restrict the geometrical limits of vision of the radiation-measuring instrument to a narrow pencil of rays along such a line of sight; and to measure the temperature, composition, and depth of flame along that same line of sight. These experimental requirements are met in the apparatus shown in complete assembly in Fig. 1. For a description of apparatus and experimental procedure sufficiently detailed to permit judgment of the validity of the results, the reader is referred to the Appendix of this paper. Briefly, equipment is provided for generating, drying, and metering carbon monoxide, for drying and metering illuminating gas, air, and oxygen, and for burning the desired mixture of gases in a Méker-type burner G (top right center of Fig. 1) consisting of an approach chamber diverging upward to a rectangular water-cooled nickel grid 15.75 in. by 0.813 in., the open length of which can be changed by slide H. To measure the temperature of the flame the method of spectral reversal of the sodium-D line is used (see Appendix). The temperature as measured is an average value along the horizontal line of sight of the spectroscope, set just to clear the tips of the small Bunsen cones of the burner grid. During the temperature measurement sodium must be supplied to the flame. This is accomplished in either of two ways, (*a*) by using compressed air in tank Q to produce mechanical atomization of a salt spray at R in the base of the burner, or (*b*) by vaporizing molten salt with an electric arc in vessel S and sweeping the vapor into burner G with a stream of nitrogen. The latter method adds no moisture to the flame. Immediately following the temperature measurement, the spectroscope system M-W-N-W'-N-L is rocked out of position on its hinged base, and replaced by a thermopile K, which consists of a single thermoelement at the focus of a gold mirror. The thermopile, containing no lenses or windows between it and the flame, is suitably diaphragmed to receive a narrow pencil of rays from the same part of the flame in which temperature was previously determined. The thermopile is calibrated by allowing it to "see" into a black-body furnace kept at a fixed and measured temperature.

Results on Carbon-Monoxide Flames

The major portion of the experimental work was conducted on carbon-monoxide flames. Radiation was measured for various burner grid lengths, fuel-gas rates, and air-gas ratios, with dry and with moist gases. In addition, oxygen was added in about half the experiments to produce higher flame temperatures or to increase the velocity of flame propagation. The range of variation of the factors studied is shown in Table 1.

TABLE 1 RANGE OF VARIATION OF FACTORS STUDIED, CARBON-MONOXIDE FLAMES

Length of burner grid, L, ft	0.164–1.312
Partial pressure carbon dioxide in combustion products, P_c, atm	0.241–0.369
LP_c, ft atm	0.047–0.476
Flame temperature, t, F	2568–3778
Radiation from flame, Btu per sq ft per hr	8540–20,070
Flame emissivity, ρ_F	0.022–0.076
Theoretical oxygen, primary, per cent	95–309
Rate of burning carbon monoxide, cu ft per hr	7.1–40.1
Ratio, oxygen to air	0–0.79

TABLE 2

Run no.	Gas rates, cu ft per hr: Carbon monoxide	Air	Oxygen	Burner length, ft	Flame temp, K	Emissivity of flame	PL (atm) (ft)
1	20.0	49.0	....	0.655	2080	0.0478	0.220
2	17.8	42.7	....	0.655	2024	0.0474	0.226
3	17.7	43.0	....	0.984	1836	0.0686	0.335
4	26.4	62.5	....	1.312	1894	0.0761	0.456
5	25.5	62.2	....	1.312	1885	0.0712	0.446
6	31.7	71.5	....	1.312	1938	0.0732	0.476
7	31.7	74.4	....	1.312	1953	0.0702	0.458
8	29.2	88.8	....	1.312	1951	0.0612	0.370
9	29.3	89.0	....	1.312	1926	0.0585	0.370
10	29.3	89.0	....	1.312	1925	0.0632	0.370
11	29.3	89.0	....	1.312	1925	0.0652	0.370
12	29.3	70.7	....	1.312	1924	0.0717	0.450
13	29.3	70.3	....	1.312	1919	0.0728	0.453
14	33.3	76.9	....	1.312	1959	0.0710	0.454
15	33.3	80.0	....	1.312	1958	0.0704	0.452

Table 2 presents the data with measured radiation converted to "flame emissivity" by dividing by the radiation from a black body at the measured flame temperature. The data are difficult to present graphically because of the number of variables. However, if the radiation from a carbon-monoxide flame is due almost wholly to radiation from hot carbon dioxide, it should depend only on the flame temperature T and on the product term P_cL, where P_c is the partial pressure in atmospheres of carbon dioxide in the combustion products and L is the flame depth or burner grid length in feet. Unfortunately the data were not taken in such a way that either T or P_cL was constant during variation of the other. However, a large number of runs were made with a small variation in the product term P_cL and a large variation in temperature. Fig. 2 shows the results for all runs in which P_cL lies between 0.16 and 0.23, with flame emissivity plotted against temperature. The data points are seen to lie definitely in a band the width of which, within the limits of experimental error of the work, is probably due to the variation in P_cL. Fig. 3 is a carbon-dioxide gas-radiation chart from Hottel and Mangelsdorf (7) based on their measurements of radiation from an electrically heated furnace full of hot carbon dioxide. Although the maximum gas temperature attainable in their experiments was about 1900 F, their extrapolation was guided somewhat by measurements of absorption from a black body at 2400 F. Values from Fig. 3 for P_cL of 0.16 and 0.23 have been converted to emissivity and superimposed on the present experimental data of Fig. 2 as solid lines. Considering the fact that these lines were obtained by an extrapolation for which the authors could not claim a probable error less than 10 per cent or perhaps 20 per cent at 3600 F, the agreement represented by Fig. 2 is excellent. It is taken by the authors as a proof that the gas-radiation chart, Fig. 3, is substantially correct in its high temperature range, along the line $P_cL = 0.2$.

The relation between P_cL and flame emissivity, for a constant temperature, is presented in Fig. 4, in which all the data of Table 2 corresponding to flame temperatures between 1800 and 2200 K have been corrected to a common temperature of 2000 K (3140 F) by assuming that Fig. 2 establishes the relation between emissivity and temperature. In Fig. 4 a solid line is superimposed from the carbon-dioxide radiation chart, Fig. 3, corresponding to 3140 F. Although the line is possibly slightly low at low values of P_cL, the agreement with the data is considered satisfactory and within the errors of individual data points. Factors contributing to the imperfection of the correlation are (a) experimental

TABLE 2 (Continued)

Run no.	Gas rates, cu ft per hr: Carbon monoxide	Air	Oxygen	Burner length, ft	Flame temp, K	Emissivity of flame	PL (atm) (ft)
16	33.4	94.0	....	1.312	1983	0.0657	0.396
17	33.4	94.0	....	1.312	1989	0.0642	0.396
18	37.2	90.0	....	1.312	2003	0.0663	0.448
19	37.1	89.5	....	1.312	2002	0.0665	0.452
20	37.1	102.0	....	1.312	2021	0.0630	0.404
21	37.1	102.0	....	1.312	2021	0.0622	0.404
22	40.1	97.5	....	1.312	2030	0.0653	0.457
23	40.1	97.8	....	1.312	2031	0.0648	0.456
24	40.1	107.0	....	1.312	2052	0.0608	0.414
25	40.1	107.0	....	1.312	2055	0.0602	0.414
26	14.0	33.2	....	0.656	1857	0.0670	0.228
27	13.9	33.2	....	0.656	1898	0.0551	0.227
28	13.9	33.0	....	0.656	1873	0.0657	0.228
29	13.8	33.0	....	0.656	1900	0.0612	0.226
30	13.8	33.0	....	0.656	1905	0.0595	0.226
31	12.0	28.5	....	0.656	1864	0.0615	0.228
32	12.1	28.5	....	0.656	1860	0.0614	0.228
33	12.1	34.9	....	0.656	1884	0.0575	0.194
34	12.0	34.9	....	0.656	1887	0.0558	0.192
35	10.0	24.0	....	0.656	1799	0.0627	0.226
36	10.0	24.0	....	0.656	1798	0.0627	0.226
37	9.9	33.3	....	0.656	1820	0.0558	0.169
38	10.0	33.6	....	0.656	1842	0.0531	0.170
39	9.9	33.4	....	0.656	1843	0.0522	0.169
40	7.2	17.6	....	0.656	1718	0.0654	0.221
41	7.1	17.3	....	0.656	1696	0.0691	0.223
42	7.1	17.3	....	0.656	1682	0.0717	0.225
43	7.2	26.1	....	0.656	1730	0.0640	0.158
44	7.2	25.8	....	0.656	1727	0.0607	0.160
45	13.8	19.7	15.5	0.164	2244	0.0250	0.0435
46	14.0	27.1	15.3	0.164	2204	0.0218	0.0464
47	13.9	27.7	15.3	0.164	2182	0.0233	0.0456
48	13.8	27.7	15.5	0.164	2206	0.0229	0.0451
49	13.8	29.3	12.1	0.164	2022	0.0297	0.0467
50	13.8	29.3	12.1	0.164	2000	0.0311	0.0467
51	13.8	29.3	12.1	0.164	1994	0.0300	0.0467
52	13.8	29.3	12.1	0.262	2163	0.0295	0.0748
53	13.8	29.3	12.1	0.262	2167	0.0294	0.0748
54	13.8	29.3	12.1	0.262	2106	0.0311	0.0748
55	13.8	29.3	12.1	0.262	2104	0.0306	0.0748
56	16.7	29.3	12.1	0.262	2330	0.0270	0.0882
57	16.7	29.3	12.1	0.262	2330	0.0266	0.0882
58	16.7	29.3	12.1	0.262	2245	0.0288	0.0882
59*M*	16.7	29.3	12.1	0.263	2244	0.0291	0.0882
60	16.7	39.0	15.4	0.525	2019	0.045	0.1396
61	16.7	39.0	15.4	0.525	2008	0.0455	0.1396
62*M*	16.7	39.0	15.4	0.525	1940	0.0442	0.1396
63*M*	16.7	39.0	15.4	0.525	1940	0.0426	0.1396
64	18.0	30.4	18.8	0.525	2049	0.0502	0.162
65	18.0	30.4	18.8	0.525	2045	0.0503	0.162
66*M*	18.0	30.4	18.8	0.525	2004	0.0456	0.162
67*M*	18.0	30.4	18.8	0.525	2004	0.0455	0.162
68*M*	18.0	30.4	18.8	0.525	1964	0.0457	0.162
69*M*	17.9	30.4	18.8	0.525	1949	0.0457	0.162
70	18.0	30.4	21.5	0.525	2022	0.0502	0.155
71	18.0	30.4	21.5	0.525	2049	0.0503	0.155
72	18.0	30.4	20.2	0.525	2078	0.0490	0.159
73	18.0	30.4	20.2	0.525	2085	0.0477	0.159
74*M*	18.0	30.4	20.2	0.525	1993	0.0454	0.159
75*M*	18.0	30.4	20.2	0.525	1986	0.0452	0.159
76	19.2	35.2	21.5	0.656	2015	0.0546	0.190
77	19.2	35.2	21.5	0.656	2005	0.0559	0.190
78*M*	19.2	35.2	21.5	0.656	1903	0.0527	0.190
79*M*	19.2	35.2	21.5	0.656	1907	0.0526	0.190
80	18.0	30.4	20.2	0.525	2032	0.0496	0.159
81*M*	18.0	30.4	20.2	0.525	2032	0.0423	0.159
82*M*	18.0	30.4	20.2	0.525	2030	0.0416	0.159
83*M*	18.0	30.4	20.2	0.525	2043	0.0475	0.159
84*M*	18.0	30.4	20.2	0.525	1999	0.0435	0.159
85*M*	18.0	30.4	20.2	0.525	2003	0.0428	0.159
86*M*	18.0	30.4	20.2	0.525	1994	0.0442	0.159
87*M*	18.0	30.4	20.2	0.525	2003	0.0431	0.159
88	18.0	30.4	20.2	0.525	2048	0.0454	0.159
89*Mb*	18.0	30.4	20.2	0.525	2052	0.0417	0.159
90*Mb*	18.0	30.4	20.2	0.525	2054	0.0427	0.159
91	18.0	30.4	20.2	0.525	2054	0.0468	0.159
92*a*	18.0	31.4	16.3	0.525	2285	0.0339	0.163
93	18.0	31.4	16.3	0.525	2298	0.0330	0.163
94*b*	18.0	31.6	16.1	0.525	2283	0.0349	0.163
95*b*	18.0	31.6	16.1	0.525	2285	0.0348	0.163
96	18.0	31.5	16.1	0.525	2306	0.032	0.17
97	18.0	28.6	15.1	0.525	2301	0.0343	0.18
98	18.0	31.5	16.1	0.525	2306	0.0318	0.17
99	21.3	39.3	17.6	0.787	2213	0.0445	0.249
100	21.3	39.3	17.6	0.787	2213	0.0442	0.249
101	22.3	43.5	22.5	1.31	1957	0.0567	0.38
102	22.3	43.5	22.5	1.31	1961	0.0564	0.38
103	13.8	20.8	10.2	0.328	2343	0.0293	0.12
104	13.8	25.0	5.5	0.328	2343	0.0264	0.159
105	13.8	25.0	5.5	0.328	2341	0.0270	0.159
106	19.1	34.3	11.3	0.655	2280	0.0377	0.227
107	19.1	34.3	11.3	0.655	2280	0.0382	0.227
108	21.8	38.3	11.3	0.655	2354	0.0352	0.234
109	21.8	38.3	11.3	0.655	2343	0.0358	0.234
110	21.8	41.4	14.4	0.983	2131	0.0477	0.322
111	21.8	41.4	14.4	0.983	2131	0.0477	0.322
112	21.8	43.5	22.5	1.31	2022	0.0513	0.37
113	21.8	43.5	22.5	1.31	1978	0.0548	0.37
114	21.8	43.5	22.5	1.31	1978	0.0548	0.37

a = All runs preceding 92 with wet salt spray.
b = Sodium in flame during radiation measurement.
M = In all runs so labeled the combustible mixture contained 0.7 to 1.8 per cent moisture.

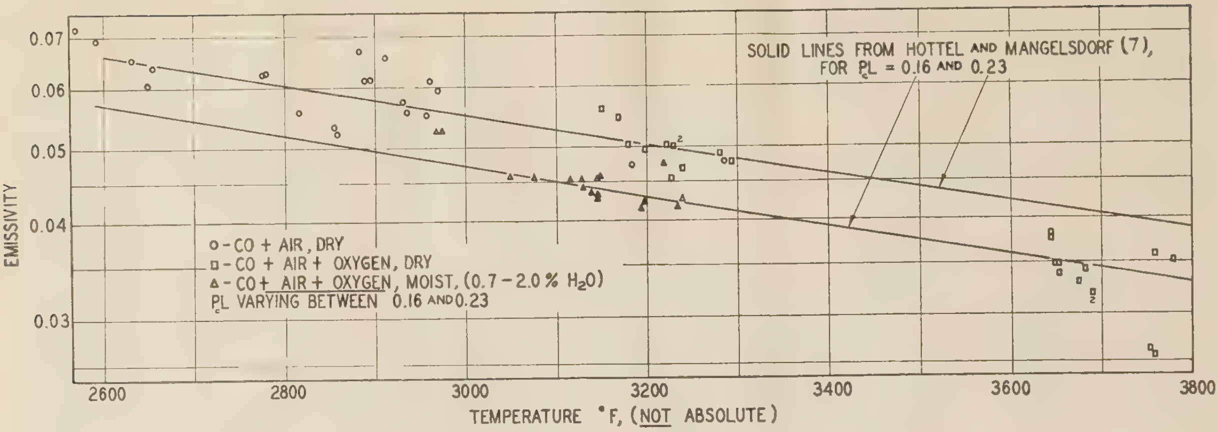
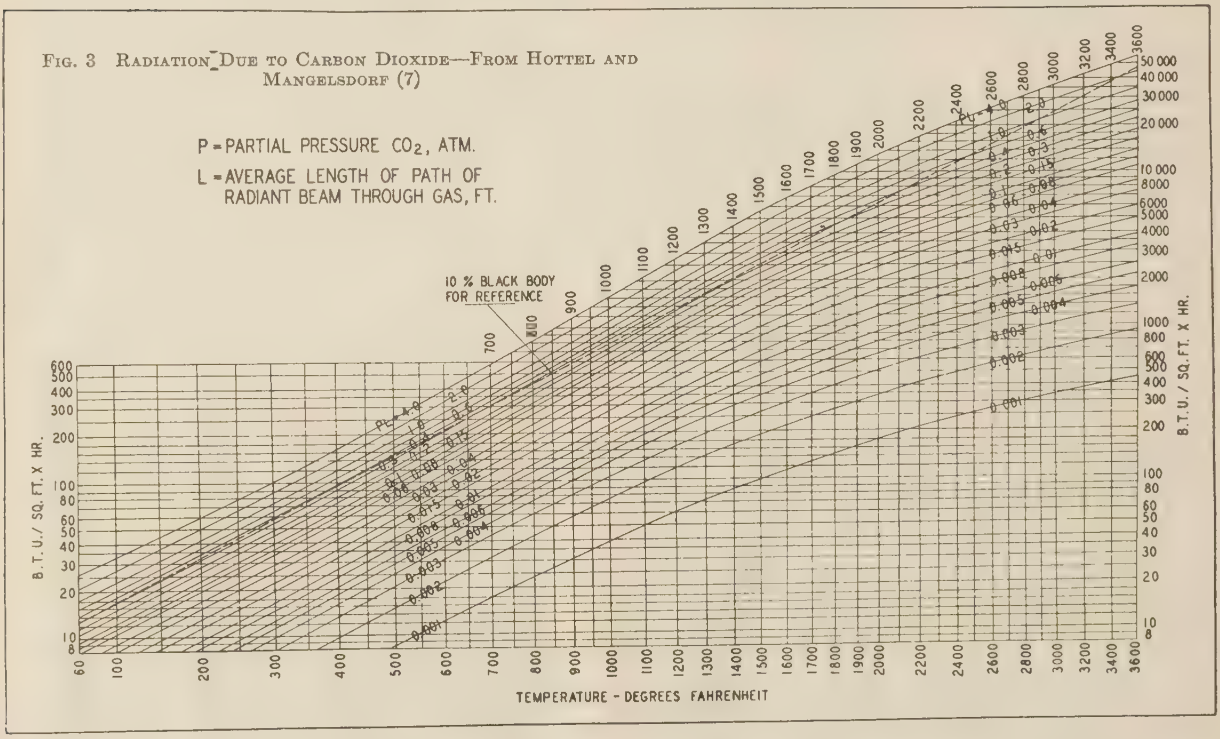


Fig. 2 Radiation From Carbon-Monoxide Flames—Effect of Temperature

Fig. 3 Radiation Due to Carbon Dioxide—From Hottel and Mangelsdorf (7)



error, (*b*) difference between actual and assumed carbon-dioxide content of the combustion products due to dissociation (estimated error 3 per cent at highest temperatures), (*c*) infiltration of secondary air, producing a change in composition of combustion products, (*d*) edge effects due to absence of abrupt change from hot flame to cold air at the near flame face.

In some of the experiments (run numbers marked *M* in Table 2) the measurements of flame temperature and radiation were made with 0.7 to 1.8 per cent moisture in the flame. The effect of the moisture, as indicated in Fig. 2, was small. Because of the effect on flame speed of the salt spray used for temperature measurements in runs 1–91 and the consequent possibility that successive measurements (first of temperature with the flame moist and then of radiation with the flame dry) did not corre-

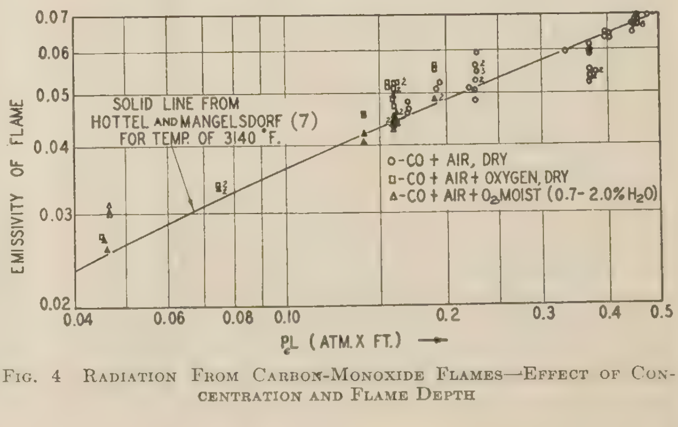


Fig. 4 Radiation From Carbon-Monoxide Flames—Effect of Concentration and Flame Depth

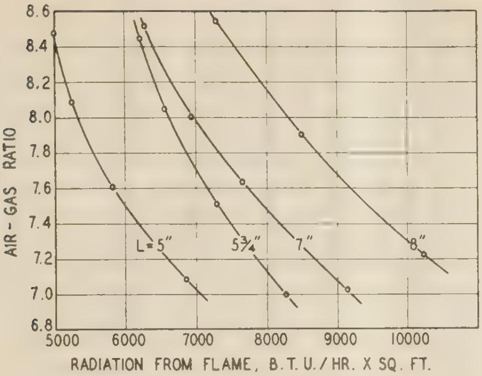


Fig. 5 Radiation From Illuminating-Gas Flames—Variation With Air-Gas Ratio and Burner Size

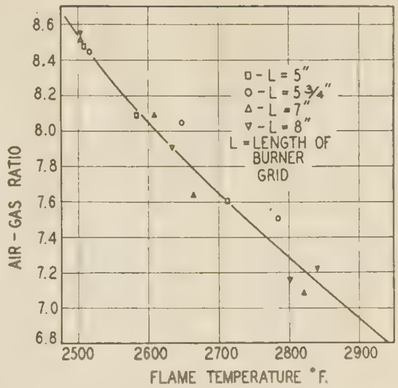


Fig. 6 Variation of Temperature of Illuminating-Gas Flames With Air-Gas Ratio

spond, the method of introducing salt was changed in all runs following No. 91. Addition of the salt in dry form had no appreciable effect on the correlation presented in Fig. 2.

Results on Illuminating-Gas Flames

With the verification of the carbon-dioxide radiation charts at high temperatures established by combustion of carbon monoxide, the next logical step is a similar study on hydrogen flames. Experimental difficulties involved in the maintenance of proper flames of hydrogen have prevented such measurements thus far. However, some results have been obtained with illuminating gas-air flames (12). Although the experiments were somewhat less extensive than those on carbon-monoxide mixtures, the data are sufficiently reliable to warrant interpretation.

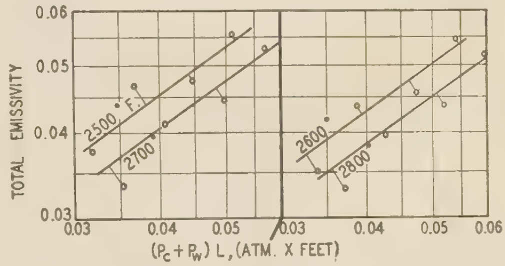


Fig. 7 Radiation From Illuminating-Gas Flames—Comparison of Experimental Results With Calculations Based on Figs. 3 and 8

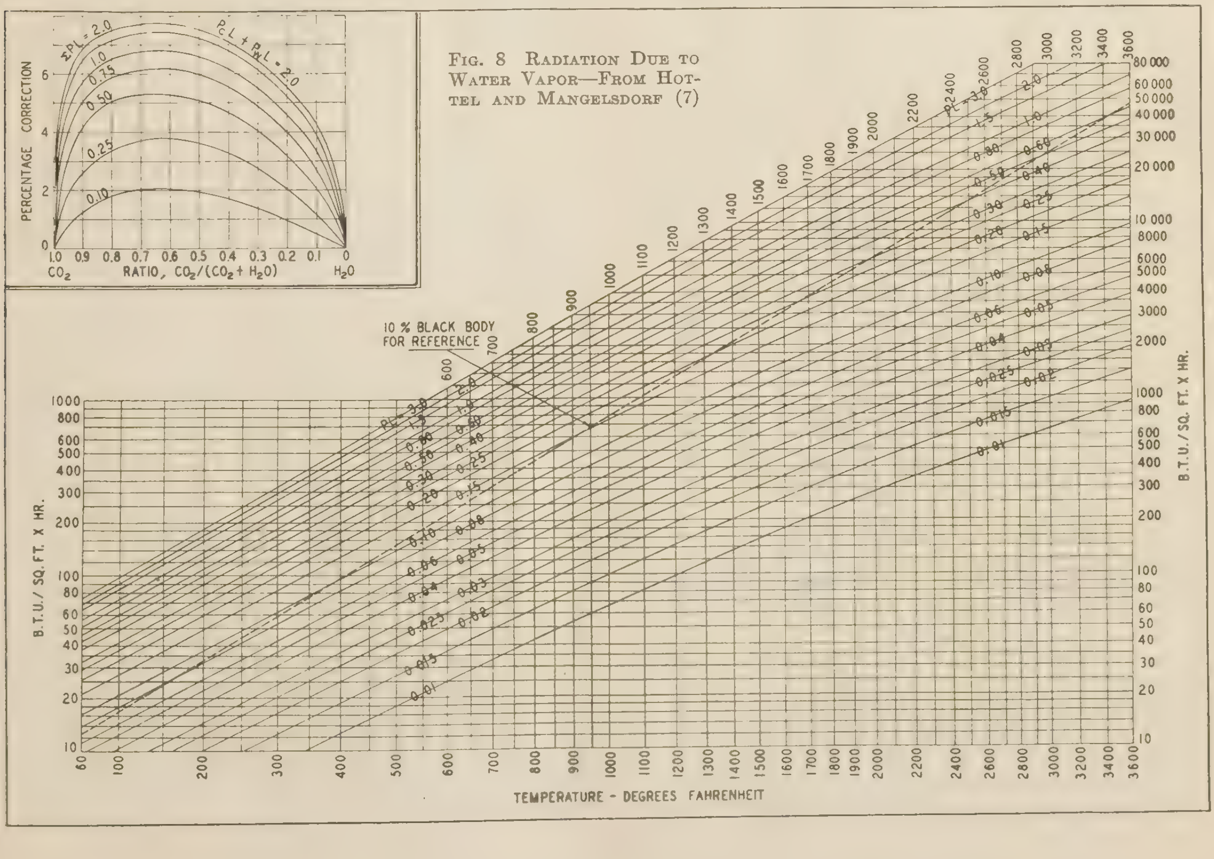


Fig. 8 Radiation Due to Water Vapor—From Hottel and Mangelsdorf (7)

Fig. 5 shows total radiation versus the air-gas ratio for four different lengths of burner grids and Fig. 6 flame temperature versus the air-gas ratio. The latter plot shows that, as expected, the measured flame temperature is independent of the length of the burner grid for the same air-gas ratio.

If as in the case with carbon monoxide, the radiation is dependent solely upon the combustion products, it may be expected that in the illuminating-gas flames the radiation will depend only upon the temperature and upon $(P_c + P_w)\,L$, where P_c and P_w are the partial pressures of carbon dioxide and water vapor, respectively, in the combustion products, and L is the length of the burner grid. Figs. 5 and 6 have accordingly been converted,

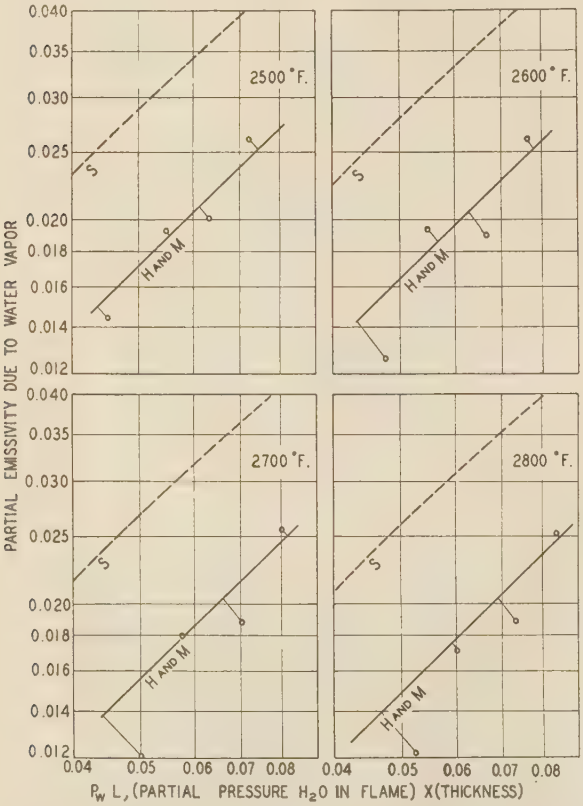


Fig. 9 Radiation Due to Water Vapor in Illuminating-Gas Flames—Comparison With Calculations Based on Fig. 8, and E. Schmidt (13)

by the use of the data of Table 3 on fuel-gas composition, to the form represented in Fig. 7; there total flame emissivity is plotted against $(P_c + P_w)L$ for each of four flame temperatures. By the use of Fig. 3 for carbon dioxide and the analogous Fig. 8 for water vapor (also from Hottel and Mangelsdorf, loc. cit.), the expected total radiation from the combustion products may be calculated. This has been done, the results appearing as solid lines in Fig. 7. It is seen that the gas-radiation charts for carbon dioxide and water vapor account satisfactorily for the measured radiation from an illuminating-gas flame.

TABLE 3 COMPOSITION OF ILLUMINATING GAS BURNED

CO_2	C_2H_4	H_2	CO	CH_4	O_2	N_2
2.2	4.2	48.0	9.4	22.0	1.2	13.0

At the time of publication of the water-vapor radiation chart, Fig. 8, it was pointed out (7) that the literature contained other data on water-vapor radiation by E. Schmidt (13), and that the new chart, though in agreement with Schmidt's results at high values of P_wL, called for considerably less radiation from water vapor at low values of P_wL. Since the first part of the present investigation has established the validity of the carbon-dioxide radiation chart at high temperatures, the data on illuminating gas may be used to obtain the radiation due to water vapor by subtracting from the experimentally measured total radiation that due to carbon dioxide. The residue, presumably due to water vapor, has been converted to an emissivity and plotted versus P_wL in Fig. 9, data points, for each of four temperatures. On that same plot appear solid lines representing water-vapor radiation, taken from Fig. 8 by Hottel and Mangelsdorf; and dotted lines representing water-vapor radiation, taken from the data of E. Schmidt. The agreement of the data points with the solid lines seems to indicate that Fig. 8 is not unduly conservative at low value of P_wL.

Conclusions

(1) The radiation from flames of carbon monoxide and illuminating gas is calculable from a knowledge of flame size, temperature, and composition.

(2) The accuracy of a chart giving radiation from carbon dioxide has been established over the temperature range 2600 to 3800 F, and the range of P_cL from 0.04 to 0.40 (atm × ft).

(3) The substantial accuracy of a chart giving radiation from water vapor has been established over the temperature range 2500 to 2800 F, at low values of P_wL where there had been some doubt as to whether the radiation chart was unduly conservative.

Acknowledgment

We wish to acknowledge our appreciation of the assistance of John E. Millard, who carried out the work on illuminating-gas flames as a part of a master's thesis at the Massachusetts Institute of Technology.

Appendix

Description of Apparatus and Procedure

Carbon-Monoxide Generator

Methods of generating carbon monoxide of high purity in large quantities have been described by Hutton and Petavel (14), Burstall and Ellis (15), and Thompson (16). The principle involved depends upon the thermal decomposition of formic acid in the presence of a dehydrating agent, according to the reaction, $HCOOH = CO + H_2O$. The generator described herein is based on the method of Thompson with a few improvements to be mentioned later.

In Fig. 1, aspirator bottle A contains 85 per cent formic acid which is fed to the bottom of the rubber-stoppered, 5-liter Pyrex reaction flask B, the rate of feed being observable through the dropping tube sealed into the supply line. Flask B is about four-fifths full of 85 per cent phosphoric acid, and is maintained at 170 C. Heat is supplied electrically at a maximum rate of 1700 watts through a coil of chromel wire held between two layers of asbestos paper, the latter being molded while wet to a snug fit around the flask. The flask is supported by its ring neck and housed in a metal container filled with 85 per cent magnesia powder. A thermocouple immersed in the phosphoric acid is connected to a pyrometer controller, giving automatic regulation of the heat supplied to the generator. The gaseous products, carbon monoxide and steam, pass through a 16 mm Pyrex tube to the water-cooled glass condenser C, also of 16 mm Pyrex. The condensed liquid escapes through a trap and the cool gas then passes up through the scrubbing tower D, filled with short sections of glass tubing, while a 5 per cent sodium-hydroxide solution from aspirator bottle E trickles down and escapes through a

trap. Rubber tubing connected to the legs of the trap permits a variable pressure up to 12 in. of water in the generator. Since this type of trap is subject to blowoff under sudden rise of pressure, a mercury safety trap F with a pilot flame at the outlet is included to regulate the maximum pressure in the generator; the mercury trap is set to blow at a pressure less than the liquid heads in the drain traps or the water-sealed gasometer when the bell has risen to its upper limit. With the exception of bottles A and E, the entire generator and the gasometer are completely enclosed under a laboratory hood.

The output of the generator is 20 cu ft of carbon monoxide per hour, being limited not by the power input, as was the case with Thompson's generator, but by the increasing tendency of formic acid to pass through undecomposed if this rate is exceeded. Analysis of the carbon monoxide leaving the scrubber showed that it contained no impurity except water vapor.

The pyrometric control of heat input at a relatively high rate and the addition of an automatic trap have produced a generator of high output that is both easier and safer to operate than those previously reported.

Burner

The burner shown at G is of rectangular shape; it has a nickel grid of the Méker type $15^3/_4$ in. long and $^{13}/_{16}$ in. wide, giving a free port area of 7.7 sq in. The casing is brazed to the sides of the grid to increase the effectiveness of the water cooling. A metal slide H, for changing the burner length, operates in slots along the top of the casing. A flared manifold distributes the gas sufficiently to give flames of uniform height. The burner inlet has been divided into two sections, all of the gas being made to pass through one or the other by the butterfly valve J. For most of the experiments, the gas was passed through the lower section for the introduction of brine spray used in measuring the flame temperature, and through the upper section during the radiation measurements, this procedure being adopted to permit radiation measurements on flames containing no sodium chloride or water vapor.

Temperature and Radiation Measurements

Spectral reversal of the sodium resonance doublet is used to measure the temperature of the gas flames. Although the accuracy of this method has been questioned lately by David (17) the objections raised seem to have been answered satisfactorily by Lewis and von Elbe (18). The continuous source L, Fig. 1, is a G.E. 4-volt, 20-amp, No. T-20, SR-8 tungsten lamp with filament 0.080 in. wide and 0.0025 in. thick; a vee is cut in one side of the filament to insure use of the same portion in all measurements. The lamp is heated by storage batteries, the current input being controlled with variable resistances, and measured accurately with the deflection-potentiometer principle of Forsythe (19) across the fixed resistance O. Images of the filament are formed at the center of the burner and at the slit of the spectroscope M by the achromatic lenses N. The spectroscope is a Gartner L250 quartz spectrograph, in which the camera is replaced by a tube for holding the eyepiece, and the Cornu prism by one of flint glass. The red brightness temperature of the tungsten lamp is measured with the precision optical pyrometer P (20). The temperatures measured in these experiments were sufficiently low to cause no rapid aging of the band filament, conveniently permitting a transfer of the optical pyrometer calibration to a temperature-current plot for the band lamp.

Two methods are used to introduce sodium into the flames. The first consists of forming a spray in the gas inlet when a small, high-velocity jet of brine from tank Q impinges against the inside of the pipe at R. A better method, in that it eliminates the undesirable effect of water vapor on both the flame speed and total radiation from the flame, is the volatilization of the salt in an electric arc between carbon electrodes immersed in a crucible of salt S. A small stream of air or nitrogen passing through drying tower U, pressure regulator T, flowmeter V, and the enclosed, water-cooled arc chamber S assists in sweeping the volatilized salt into the burner manifold, as well as preventing flow of the gas mixture into the arc chamber with resulting ignition. If air is passed through the arc a correction is made for CO_2 formed at the electrodes.

The radiant energy emitted by the flames is measured with a reflecting-type, total-radiation thermopile K, focused on the center of the flame. The latter consists of a single spot-welded junction of copper and constantan wires, 0.001 in. in diameter, the free ends of which are fastened into two slotted copper leads cemented firmly in a celluloid block $^9/_{16}$ in. in diameter. These leads also form the cold junction. The target is 0.050 in. in diameter of tinfoil 0.001 in. thick, thoroughly coated with lampblack on the front, and cemented in close contact with the junction on the back surface. The mirror is a bright, opaque film of gold sputtered on the concave front surface of quartz; it is $^3/_4$ in. in diameter and has a focal length of 5 in. The thermopile, mounted in a telescope for focusing, is housed with the mirror in a wooden box K to which a water-cooled shield and shutter are fastened. Trays of soda-lime and calcium chloride are placed on the bottom of the box, and a slow stream of air, dry and free of CO_2, is passed through the box during measurements.

A Leeds and Northrup type-HS reflecting galvanometer, mounted on a Julius suspension and the whole enclosed in a compartment to protect it from air currents, is connected to the thermopile leads. The scale, placed at 3.5 m, is readable to 0.1 mm through a high-magnification telescope; the sensitivity of the galvanometer viewed at this distance is 101 cm deflection per microampere.

Two black bodies X of the type described by Coblentz (21), the temperatures of which are measured by carefully calibrated thermocouples, are used to calibrate the thermopile. These bodies are held at constant temperatures of about 730 F and 1820 F; over the thirteenfold range of variation of radiation intensity represented by these two radiation standards, the calibration constant of the thermopile-galvanometer system varied no more than 3 per cent, and the average was much less. A calibration is made before and after each group of flame experiments, the arithmetic mean of the constants so obtained being used for the calculations in the group.

In view of the marked dependence of radiation upon temperature, special experimental care is required to make the two properly related, particularly in a flame where temperature gradients may be great. Therefore, the band lamp, lenses, and spectroscope are mounted on an optical bench which is supported by two sturdy rods fastened into a steel base; the base is fastened to the desk top with hinges so that this system or the thermopile can be placed in line with the burner. Once in correct alignment and clamped, the separate parts undergo no relative motion and no further adjustment is required. The thermopile housing is mounted similarly on a solid base which slides back and forth transversely between fixed stops on a metal track. Once properly aligned and the stop fastened, the thermopile can also be moved and returned to the correct position.

In order to minimize the effect of temperature gradients, the flames are made as large as possible and the distance from the burner grid to the optical axis is chosen so as to place the latter in the most uniform temperature range. In addition, diaphragm W limits the cone of rays viewed during the temperature measurements, and is of such size that any ray drawn from the eyepiece image through the optical system falls well within lens N_2. The limitation of field of view of the thermopile system is accomplished

by two diaphragms of 0.125 and 0.030 in. in diameter, placed immediately in front of the mirror and the target, respectively, in the thermopile housing. The pencil of rays through the flame is thereby restricted to a diameter of 0.15 in. at the image plane near the center of the burner grid, and a maximum of 0.23 in. at the back edge of the grid.

Control of Gas Mixture

In order to avoid variation in mixture ratio while radiation and temperature measurements are being made, a $^3/_4$-in. type-*A* Reliance pressure regulator is placed in the carbon-monoxide line, and $^3/_4$-in. type-*H* regulators are placed in the other gas lines. The rates of flow are measured on calibrated orifice meters *V*. In almost all cases the total rate is such that the flame is just beginning to lift away from the grid. The gases, dried in the calcium-chloride towers if required, are tested for water vapor by aspirating a stream of gas from the manifold sampling tube through a dew-point apparatus.

Procedure

The thermopile is calibrated against the black bodies previously heated to constant temperatures of about 750 F and 1800 F; the carbon-monoxide generator, already up to operating temperature, is started; the mixture ratio is adjusted to give the kind of flame desired; the dew point of the mixture is determined; the thermopile is brought into position and galvanometer readings taken for about 15 successive 15-sec periods with the shutter alternately opened and closed; the thermopile is pushed aside; with the lamp already hot the optical bench is swung into position, the sodium generator is started, and three or four reversals made in the spectroscope, the potentiometer being read for each. This procedure, usually repeated as a check, is followed for each mixture ratio and burner length.

Calculations

The recorded brightness temperatures at $\lambda = 0.665\mu$, without the lens interposed, are corrected to those at $\lambda = 0.589\mu$, with the lens, by substituting appropriate values in Equation [1].

$$\frac{1}{T_R} - \frac{1}{T_{y,L}} = \frac{0.589 \ln ABp_Y - 0.665 \ln Ap_R}{14{,}330} \quad \ldots\ldots [1]$$

where T_R = brightness temp, deg K, at $\lambda = 0.665\mu$, viewed through the bulb but without a lens

$T_{y,L}$ = brightness temperature, deg K, at $\lambda = 0.589\mu$, viewed through the bulb and lens

A = transmissivity of the lamp bulb

B = transmissivity of the lens

p_R = emissivity of tungsten at $\lambda = 0.665\mu$ and true filament temperature, obtainable from T_R (22)

p_Y = emissivity of tungsten at $\lambda = 0.589\mu$ and true filament temperature.

The transmissivity of the lens is found experimentally by measuring the red-brightness temperature of the band lamp, with and without the lens between it and the optical pyrometer. For a fixed lamp temperature

$$\ln B = -\frac{14{,}330}{0.665}\frac{(T_1 - T_2)}{(T_1T_2)} \quad \ldots\ldots\ldots\ldots [2]$$

where B = the transmissivity of the lens

T_1 = red brightness temperature of lamp, deg K, without lens

T_2 = red brightness temperature of lamp, deg K, with lens

Substitution into [2] of corresponding values obtained at several temperatures give an average value of 0.92 for *B*. This is adopted also for the transmissivity *A* of the lamp bulb and is used throughout for calculating the correction.

Values of p_R and p_y are taken from the data of Forsythe and Worthing (22).

The correction varies from +7 C at 1670 K to +14.6 C at 2200 K.

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Safe Operation of High-Speed Locomotives

By B. S. Cain,[1] Erie, Pa.

In this paper are presented the basic theories, test information, and the chief conclusions of an investigation of factors entering into the mechanical design of locomotives for high-speed operation. The author points out that (1) recent trends in railroad operation have been toward higher speeds over longer distances and (2) the development of electric and oil-electric power, and light-weight equipment have changed many of the conditions under which these higher speeds are attained. In view of the radically new types of equipment available, the increased severity of requirements both as to performance and cost of equipment, and the necessity for preserving the railroad's record for safety, it has been necessary to make an extended theoretical and experimental analysis of high-speed locomotive operation. Such an analysis is given in this paper.

The fundamental problem in regard to the safe operation of high-speed locomotives has been to explain, and to learn to control by design means, the oscillations of locomotives which develop at high speeds, and which place a definite limit on the safe speed at which any particular design of locomotive can be operated over a given section of track. These oscillations consist partly of nosing and rolling motions produced by the sidewise components of the wheel-friction and flange-impact forces when the wheels are out of exact line with the track. Instability occurs when these forces increase the energy of oscillation faster than the damping forces can reduce it. Other oscillations are due to rough track. It is the problem of the designer to control the design, the wheel arrangement, the spring constants, the weight distribution, and the clearances so as to minimize the tendency to instability and to obtain ample margins of safety under varying conditions of track.

Testing of locomotives has been improved during recent years. Stresses in the track can be accurately measured with electric strain gages. The author has used these gages extensively to make continuous records of lateral pressures between locomotive axles and frames on a number of locomotives up to very high speeds. The development of theories to determine the stresses and motions of locomotives at high speeds has proceeded before and coincidently with the test program. The theory as now developed logically explains the test results in so far as the fundamentals are concerned. Some of the principal results of this work are outlined in this paper, reference being made particularly to the high-speed electric locomotives on which the most complete experimental information exists.

General Theory

It is a matter of frequent observation that high-speed equipment may oscillate continuously of its own accord. The only source of energy available to keep these oscillations going is the driving energy which maintains the speed. In a balanced machine, the only way in which this energy can be tapped to produce a sustained oscillation is through friction at the tread of the wheels. Whenever the locomotive wheel base makes a small angle with the track, friction makes it run sideways until the wheel flanges hit the rail. As a result of the impact, the wheel base may become pointed toward the other rail. If this motion persists or increases there is an ever-present tendency to breakage or derailment.

A study of the friction at the head of the rail shows that for oscillations at high speed the frictional force is not constant, but actually approaches a creepage force, varying with the amount of slip. Theoretical calculations which have been made of this creepage force are fairly representative of test data.

The initial motion which starts an oscillation may be due either to inherent instability of the wheel base or to rough track. Both cases have been reduced to calculations by which the tendency of locomotives to start oscillation may be compared. The calculation of the oscillation itself has been divided into two parts.

First, assuming some reasonable type of oscillation, it is possible to calculate the speed at which the oscillation will just sustain itself because the losses and gains of energy are equal. Further details of these calculations are given in the Appendix. This speed is proportional to the frequency of the oscillation, which is still unknown. Other factors, such as the lag of the rear end behind the front end in side motions, can be calculated to give the lowest critical speed.

After this energy calculation is made, a study of the motion in more detail is made to determine the frequency and to ascertain whether the forces resulting from such a motion will approximate those required to produce it. The calculation is essentially one of successive approximations. It is practical because the frequency is not affected much by the friction and damping. Therefore, the calculation of frequency can, after some experience, be made quite simply in many cases.

These calculations apply chiefly to straight track. On curves, it is usually sufficient to draw a diagram of the locomotive in the curve showing the locomotive in equilibrium and to see whether, with large variations in friction, the position is stable. The author has found it most useful to use a diagram in which a reduced scale is used for the direction corresponding approximately to the length of the locomotive and to use a full scale at right angles to this direction. Then clearances are approximately full size and the diagram is accurate at all points.

These calculations, particularly the energy method of studying oscillations on straight track and the layout method for curves, generally allow a steady locomotive to be designed or indicate the reason for observed instability. If severe flange pressures cannot be avoided, their values can be estimated by following these methods further. The allowable limits of pressure have been investigated in an elementary way to bring out the important factors, and some methods of calculation have been outlined, relat-

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ing to derailment and lateral shifting of track. These methods of calculation are given in the Appendix.

Test Methods

Most of the author's tests have been made to record lateral pressure between the axles and the locomotive frame. This is done by modifying the journal boxes so that the axles are restrained laterally by steel "weigh-bars" one on each side, these bars being adjustable to give the required clearances. Lateral pressure bends one of these bars, and an electromagnetic strain gage on the bar records the pressure on an oscillograph record. The bars are calibrated statically in a testing machine. Records have been obtained both on films and on paper rolls 100 ft long. Pressures on all axles of a locomotive may be recorded simultaneously.

Tests have also been made in which truck motion, acceleration, rail stress, and rail pressure have been recorded. The author has also had access to numerous tests of this nature made by other investigators in this field.

Conclusions

In order to keep a locomotive of conventional type from exerting high flange pressures on straight track when moving at high speed, the wheel base should be long and the clearances small to reduce angularity in the track. The independent swiveling of trucks on straight track should be prevented. Wheels near the center of the locomotive should be flanged and should preferably have lateral freedom in the frame, so that they guide themselves and do not reduce the stability of the wheel base.

Springs should be as soft as is practical to cushion blows and reduce the forces due to oscillation.

On high-speed curves, truck restraints and wheel bases should be studied to avoid instability. On sharp curves the long wheel base should be able to adjust itself to reduce angularity between flanges and rail, and to avoid high flange pressures.

The height of the center of gravity of the locomotive is a compromise. In general the height should be reduced where the locomotive is well guided and where the track is good.

Excessive axle weights should be avoided. In actual designs, these ideals can only be approached. Long rigid-frame locomotives tend to oscillate on high-speed curves and to derail on sharp curves. Therefore, designs must aim particularly at reducing these tendencies. Because articulated locomotives tend to oscillate on straight track at high speed, restraining devices are used to eliminate this tendency. Truck restraints with decreasing characteristics may also be necessary to avoid excessive flange pressures on sharp curves.

Good track maintenance and uniform construction and stiffness with the minimum practical flange clearance are always of great importance.

It is very desirable to obtain more measurements of forces produced by different locomotives and particularly to make further tests on the limiting forces which track can safely withstand.

An energy method of calculating locomotive oscillations has been developed by B. F. Langer and J. P. Shamberger,[2] but their calculations do not agree with those given in this paper.

Acknowledgment

The author is indebted to many engineers for information and suggestions. Particular reference should be made to E. G. Keller who, while with the General Electric Company in 1931 and 1932, cooperated with the author in devising methods for detailed calculation of locomotive oscillations. Dr. Keller worked out these methods in numerical detail, thereby providing clear representations of locomotive behavior and suggesting practical approximate methods. The author is indebted to F. M. Graham for information on track and for results of tests on which the theory of track movement is based. R. Eksergian and P. L. Alger have made valuable suggestions in regard to the subject matter and presentation of the paper.

[2] "Lateral Oscillations of Rail Vehicles," by B. F. Langer and J. P. Shamberger, A.S.M.E. Trans., vol. 57, 1935, paper RR-57-4.

Appendix

Friction and Creepage Between Wheels and Rails

According to a common theory, a wheel will roll on a rail without slipping until the tangential force at the point of contact is sufficient to overcome the friction, given by the load on the rail multiplied by the coefficient of friction. As long as the force retains this value, the wheel will continue to slip. The coefficient varies greatly with the conditions. Usual typical values are given in Table 1.

TABLE 1 COEFFICIENTS OF FRICTION FOR DIFFERENT RAIL CONDITIONS

Condition	Coefficient
Good dry conditions, no slipping	0.35
Wet surfaces, no slipping	0.25
Dry surfaces, slow speed and slight slipping	0.25
Dry surfaces, high speed and slight slipping	0.15
Dry surfaces, wheel spinning	0.05

If the wheel and rail were rubber, the rolling wheel could creep in the direction of any tangential force, due to the elasticity of the materials in contact. Steel being elastic, the same creep must occur to a reduced extent and theoretical studies indicate that the amount of the creep is sufficient to allow many of the

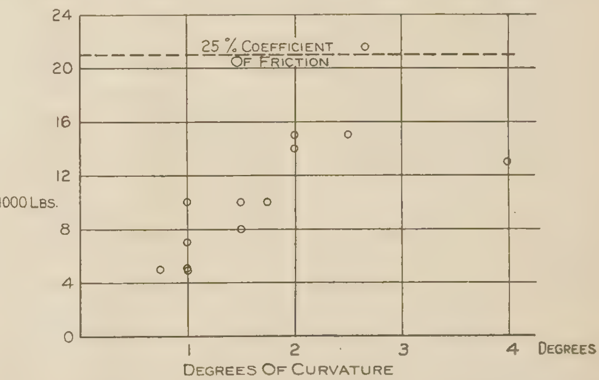

Fig. 1 Lateral Force Required to Slip a Locomotive Axle Laterally as Obtained From Tests at Speeds From 25 to 60 Mph

motions of actual locomotive wheels without requiring the whole area of contact to slip. The creepage is defined as the ratio of creeping to rolling displacement of the wheel. The creepage coefficient is the ratio of tangential force at the point of contact to the creepage.

The creepage coefficient has been calculated for a particular case of F. W. Carter,[3,4] who gives the formula

$$f = \text{creepage coefficient, lb}$$
$$= 3500\,(\text{wheel diameter, in.} \times \text{axle load, lb})^{0.5}$$

The present author has made about 50 measurements of the lateral force required to hold an axle with blind tires while going

[3] "On the Action of a Locomotive Driving Wheel," by F. W. Carter, Proceedings, Royal Society of London, series A, vol. 112, 1926, p. 151.

[4] "The Running of Locomotives With Reference to Their Tendencies to Derail," by F. W. Carter, Selected Engineering Paper No. 91, The Institution of Civil Engineers, 1930.

around various curves. The results are rough, due to such conditions as fluctuations in the force and uncertainty as to superelevation, but the results plotted in Fig. 1 for speeds of 25 to 60 mph and in Fig. 2 for speeds of 60 to 70 mph show a marked tendency for the lateral force to decrease to zero with the curvature, for curves less than about 3 deg. The angularity of the axle to the track radius can be estimated for the locomotive on which these tests were made. It increases roughly with the curvature. Calculations in the range from 1 to 2 deg give an approximate angularity (and therefore creepage) of from 0.0007 to 0.0017 and an average creepage coefficient of about 8,500,000 lb. Dr. Carter's[3] formula gives a creepage coefficient in this particular case of 8,600,000 lb. The difference is less than the probable error of either figure.

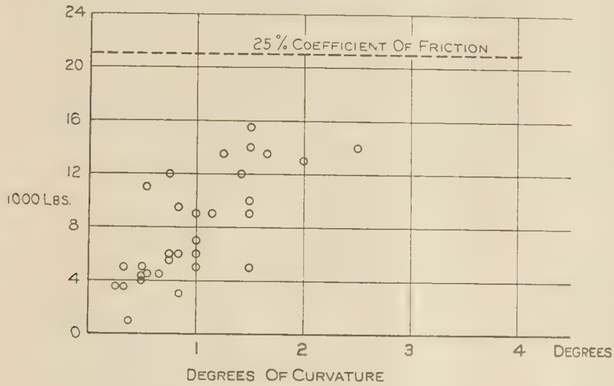

Fig. 2 Lateral Force Required to Slip a Locomotive Axle Laterally as Obtained From Tests at Speeds From 60 to 70 Mph

The creepages required to allow the oscillation of locomotives at high speeds on tangent track are within the range of these tests and the author therefore believes that the creepage theory is a legitimate one to use in such studies.

It should be noted that J. Buchli[5] made model tests and concluded that: "To cause lateral movement of a hauled wheel from its neutral position, a force of definite magnitude is necessary, the value of which is equal to the product of the load on the wheel multiplied by the coefficient of friction. . . . "

Applying Carter's formula to the loaded wheel used by Buchli (about 12-in. diameter with less than 1000 lb load) the maximum creep before complete slippage would be of the order of 0.04 per cent, which would be inappreciable in the test. Buchli's result applies to wheels running with considerable angularity on sharp curves and does not contradict the author's observations at high speed, where the angularity of wheels is very small.

Oscillations Started by Inherent Instability

There are some locomotives which are unstable on perfect track, even before the flange forces come into play. Such a locomotive will always start to deviate from steady motion down the track. When the flanges hit the rail, the resulting oscillation may sustain itself or may decrease, but if it decreases a new one will start, resulting in more flange impacts and so on indefinitely. This natural instability has been studied by F. W. Carter.[3,4] Broadly there are two causes of instability.

The first is the tendency of certain combinations of wheel bases to get out of line. A 4-4-0 locomotive with sufficient guiding force in the engine truck will follow that truck and tend to straighten itself. The same locomotive reversed, 0-4-4, will buckle because the rear truck tends to run in its own direction, which causes the drivers to become more and more out of line with the truck. A stable locomotive, from this point of view, will tend to straighten its wheel base and proceed in a straight line, but this line will not necessarily be the center line of the track. The angular deviation will be that produced by an accidental disturbance and will not be increased by buckling of the wheel base.

The second cause of instability is the coning of wheel treads. A rigid truck wheel base with all wheels equal and coned, moving slowly down uniform track will, if given a slight displacement, tend to move sinuously, the motion repeating after a certain distance. As the speed increases, the motion becomes unstable, due to the mass and moment of inertia of the truck. If the wheels are of different diameters, a truck with smaller wheels leading tends to run more stably and one with smaller wheels trailing tends to run more unstably.

In the author's opinion the effects of coning on large locomotives are generally small, because (1) the coning of new wheels is generally small, (2) the coning of new wheels is quickly worn down, and (3) the clearances between axles and truck frames usually tend to reduce its effect. On the trucks of light-weight high-speed cars, the effect of coning is considerable.

Oscillation Started by Rough Track

If track is poorly surfaced it will start a motion of the locomotive. A rolling motion results from variations in the level of the two rails. In general, this roll is reduced by giving the engine a long natural period of roll on its spring system and by increasing the locomotive length, so that the various axles produce their effects at different times and interfere with each other. If the surface irregularity is very short, the effect on each axle is to produce an angular velocity

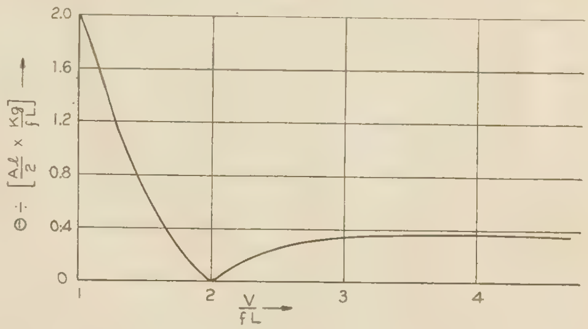

Fig. 3 Angle of Roll of a Vehicle After Passing Over a Track Irregularity

$$(Al/2) \times (kg/I) \times (1/V)$$

Where A = maximum cross-tilt of the track at the irregularity; l = the length of the irregularity which is assumed to have the shape, $A[1 - \cos(2\pi x/l)]$; k = moment acting on the body of the locomotive, due to the springs, per unit angular movement of the axle; g = acceleration due to gravity; I = the moment of inertia of the locomotive body in roll about an axis through the centers of the axles; and V = the speed of the locomotive. The angular velocity can be represented by a vector rotating with angular velocity equal to 2π times the natural frequency of the locomotive in roll.

If the next axle is a distance L behind the first one, it will receive its impulse after a time L/V and the subsequent motion is represented by adding to the first vector a second of a magnitude calculated for the second axle and lagging a time L/V

[5] "Behavior and Movement of Locomotive Wheels on the Track," by J. Buchli, Bulletin of the International Railway Congress, new series, vol. 6, no. 6, June, 1924, pp. 417–437.

behind the first. The effects of successive axles can be added in a similar way. If the locomotive springs are heavily damped, the resultant vector can have this damping applied to it during its rotation.

This vector diagram will show the rolling motion completely. It will be clear from inspection that if a locomotive has n similar axles which are equally spaced, it will leave the irregularity with no roll at all if the time taken to pass over it is just equal to $(n-1)/n$ times the natural period of the locomotive in roll (assuming undamped springs). Fig. 3 shows the relation between the final roll and the speed of a symmetrical two-axle truck plotted in dimensionless coordinates. As V/fL increases above 2, the roll

$$\theta/[(Al/2) \times (kg/fL)]$$

increases to about 0.36 and then decreases steadily toward zero.

Roll in itself will not always start any other oscillation, but it constitutes a reservoir of stored energy which produces nosing if any accidental angularity occurs.

Flange Impacts

When a wheel flange runs against the rail, its lateral motion is quickly stopped. The locomotive begins to roll and the flange tends to bounce back away from the rail. If the center of gravity of the locomotive continues to move in the same direction after the impact, the springs, in bringing the locomotive back from its roll, will pull the flange back again and a second impact will take place. On the other hand, if the center of gravity is low, so that the impact reverses the direction of its motion, the flange may not come in contact with the rail again until the whole locomotive has completed another oscillation. Actually, the impact is reduced by the effect of the increasing diameter in the throat of the wheel flange, which tends to prevent direct impact on the flange. The detail calculations of multiple flange impact are not difficult, but may be very laborious because of the repeated discontinuities in the forces. In many cases where the locomotive as a whole is being studied it will be found convenient to make the following simplifying assumptions:

With the flange hitting the rail, consider a point above the axle situated on the longitudinal axis through the center of gravity of the locomotive body. If, after impact, this point moves away from the impact point, then subsequent impacts are neglected. If this point moves toward the impact point, then the impact is recalculated, assuming that the flange is guided along the rail smoothly as the result of an inelastic impact. For the dividing line, where the point is stationary, the two assumptions do not give the same result, but the actual motion is considerably affected by small variations, so that it is not unreasonable to have a case where the solution is uncertain between two limits. When spring friction or damping is taken into account, the impacts are usually simplified, the approximations become more accurate and agree well among themselves. If necessary, the more exact calculations of multiple impact can always be made.

The theory of multiple impact is borne out by the nature of actual blows as recorded in high-speed oscillograms. Under varying conditions, particularly of spring friction, these show (1) single blows, (2) fluctuating blows, and (3) blows in which the rebound actually divides the blow into two parts, an initial peak, falling to zero and followed by a fluctuating pressure of longer duration. Calculation of impact for a single unit is made as follows:

If h = the height of the center of gravity above the centers of the axles, k_x = radius of gyration about a longitudinal axis, k_z = radius of gyration about a vertical axis, l = distance of impact ahead of the center of gravity, W = weight of the body, u = lateral velocity of the center of gravity, ω_z = angular velocity of the nosing action, and ω_x = angular velocity of roll, then the equivalent mass at the point of impact is

$$\frac{W}{g}\left[\frac{1}{1+(h^2/k_x^2)+(l^2/k_z^2)}\right]$$

and the impact is

$$\frac{W}{g}\left[\frac{2}{1+(h^2/k_x^2)+(l^2/k_z^2)}\right](u+l\omega_z-h\omega_x)$$

Therefore, after the impact, the velocity of the point $x = l$, $y = 0$, and $z = 0$ is

$$\left[u-\frac{2(u+l\omega_z-h\omega_x)}{1+(h^2/k_x^2)+(l^2/k_z^2)}\right] + l\left[\omega_z-\frac{2l}{k_z^2}\left\{\frac{u+l\omega_z-h\omega_x}{1+(h^2/k_x^2)+(l^2/k_z^2)}\right\}\right]$$

$$= u+l\omega_z-\left\{\frac{2[1+(l^2/k_z^2)]\,(u+l\omega_z-h\omega_x)}{1+(h^2/k_x^2)+(l^2/k_z^2)}\right\}$$

Critical Speed on Tangent Track

The energy method of calculating critical speed is similar to other energy methods used in engineering calculations. A reasonable sinuous motion is first assumed and the loss or gain of energy of the locomotive is calculated for this motion. The loss and gain depend on the speed of the locomotive, and the lowest speed at which they balance is called the critical speed. Subject to reasonable limitations the shape of the assumed path can be varied to obtain a minimum critical speed.

Assuming a constant forward speed V, the locomotive can dissipate energy (*a*) by lateral and angular oscillations which cause the wheel treads to slip or creep on the rail, (*b*) by rolling, due to friction and damping in the springs, (*c*) by the operation of such devices as internal friction plates and dampers, (*d*) by impacts between flanges and rails, and (*e*) by rough track where this reduces existing oscillations. It can gain energy by (*f*) the tendency of any angularity of the wheel base to give the engine a lateral velocity and (*g*) rough track where this starts or increases oscillations. It is easier to consider first (*a*) and (*f*), the forces at the head of the rail. It is clear that with perfect uniform track it is not possible to gain or lose energy in flange impacts because these are uniformly elastic and the only forces left are those of slip or creep between the head of the rail and the tread of the wheels.

The calculation is based on the fact that the rate of gain of energy at each point of contact is equal to the force acting on the wheel multiplied by the velocity of motion of the point of application in the direction of the force. If the motion is in a direction opposite to the force, the gain becomes a loss of energy. First assume that the wheels creep, with a creepage coefficient f.

Let an arbitrary reference point o shown in Fig. 4, be taken which moves down the center of the track with velocity V. Let the wheel base be deflected a distance y to o' and rotated an amount θ. Then the lateral deviation of an axle distant l ahead of the reference point is $y + l\theta$. The lateral velocity is

$$\dot{y}+l\dot{\theta}$$

Of this velocity, $V\theta$ is due to rolling displacement of the wheel, and the velocity of creep is

$$\dot{y}+l\dot{\theta}-V\theta$$

The creepage force is

$$-(f/V)\,(\dot{y}+l\dot{\theta}-V\theta)$$

The rate of gain of energy is creepage force times lateral velocity

$$= -(f/V)\,(\dot{y} + l\dot{\theta} - V\theta)\,(\dot{y} + l\dot{\theta})$$

Now assume that y and θ are sinusoidal, which is a reasonable first approximation, and let $\omega = 2\pi f$, where f is the frequency. Then

$$y = Y \sin \omega t$$

θ may be written $\theta \cos(\omega t + \epsilon)$, θ being now the maximum value, and the average rate of gain of energy

$$= -(f/V)[(Y^2\omega^2/2) + Yl\theta\omega^2 \sin\epsilon + (l^2\omega^2\theta^2/2)] + f(\theta Y\omega \cos\epsilon/2)$$

For the whole wheel base, summing over various axles

$$\text{Gain of energy} = -\Sigma(f\omega^2/2V)[Y^2 + 2lY\theta \sin\epsilon + l^2\theta^2 - (V/\omega)Y\theta\cos\epsilon]$$

For the critical speed, this must be zero, or

$$V_{cr} = \omega\left[\frac{\Sigma f(Y^2 + 2lY\theta\sin\epsilon + l^2\theta^2)}{\Sigma fY\theta\cos\epsilon}\right]$$

$$= \omega\left[\frac{Y}{\theta\cos\epsilon} + \frac{2\Sigma fl}{\Sigma f}\tan\epsilon + \frac{\Sigma fl^2}{\Sigma f}\,\frac{\theta}{Y\cos\epsilon}\right]$$

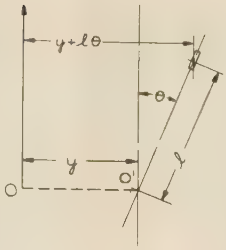

FIG. 4a POSITION DIAGRAM FOR A SIMPLIFIED TRUCK WHEEL

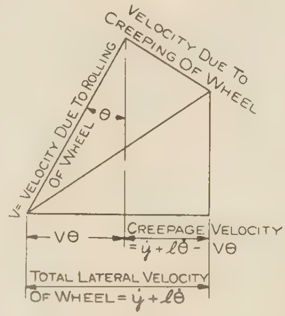

FIG. 4b VELOCITY DIAGRAM FOR A SIMPLIFIED TRUCK WHEEL

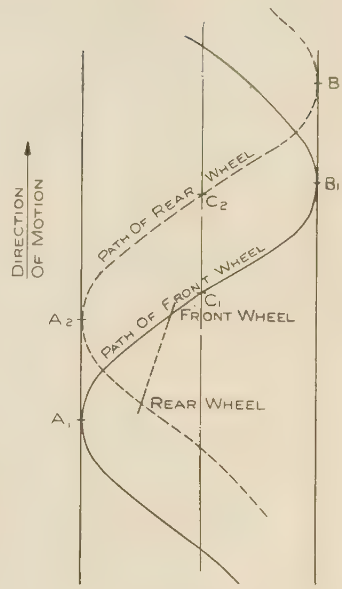

FIG. 5 DIAGRAM OF MOTION OF A SIMPLIFIED TRUCK

Choose the origin of the coordinates so the $\Sigma fl = 0$. This makes the second term vanish, making it obvious that the least value of critical velocity V_{cr} is given by the $\cos\epsilon = 1$, so that

$$V_{cr} = \omega[(Y/\theta) + (\Sigma fl^2/\Sigma f)(\theta/Y)]$$

which is a minimum when

$$Y/\theta = (\Sigma fl^2/\Sigma f)(\theta/Y)$$

Therefore

$$Y/\theta = \sqrt{(\Sigma fl^2/\Sigma f)}$$

giving

$$V_{cr} = 2\omega\sqrt{(\Sigma fl^2/\Sigma f)}$$

This calculation neglects the effect of the track gage, although it can be taken into account by adding the effect of longitudinal forces due to the different longitudinal speed of the two wheels on one axle.

The rate of gain of energy is increased by

$$-0.25(f/V)S^2\dot{\theta}^2$$

where S = the track gage measured between center lines of the tread contacts. Therefore, the speed

$$V = 2\omega\sqrt{\frac{\Sigma f[l^2 + (S^2/4)]}{\Sigma f}}$$

Finally, the effect of lateral clearances in the locomotive has to be taken into account. This theory is a first approximation, the motion of all parts being assumed sinusoidal. It follows that a proper assumption to make with regard to axles with clearance in the journal boxes is to take their motion as sinusoidal and to make the assumed positions coincide with the real positions at the extreme points of the cycle, where impacts take place. Therefore, assume that the motion of the frame laterally with respect to the axle is proportional to the motion of the axle laterally with respect to the track and that the internal clearance is taken up at the same time as the track clearance. In any oscillatory motion this will give very closely the same extreme conditions, the same track impacts and the same net energy transfer at the head of the rail as in the real motion.

As a second approximation, if necessary, the motion calculated in this way can be plotted and corrections made graphically.

Let C_i = the internal clearance between axle and frame, C_t = the track clearance between flanges and rail, and μ = the ratio of track clearance to total clearance, so that

$$\mu = C_t/(C_i + C_t)$$

First assume that μ is the same for all axles. Then

$$V_{cr} = 2\mu\omega\sqrt{\frac{\Sigma f[l^2 + (S^2/4\mu^2)]}{\Sigma f}} = 2\omega\sqrt{\frac{\Sigma f[\mu^2 l^2 + (S^2/4)]}{\Sigma f}}$$

so that internal clearance is equivalent to a reduction in wheel base.

If different axles have different clearances, so that each axle has its own value of μ, then the origin of the coordinates is to be taken so that $\Sigma f\mu^2 l = 0$ and

$$V_{cr} = 2\omega\sqrt{\frac{\Sigma f\mu^2 \times \Sigma f[\mu^2 l^2 + (S^2/4)]}{(\Sigma f\mu)^2}}$$

Losses due to spring friction and the effects of coning may be included similarly. These formulas depend on the creepage theory of wheel action. It is worth while to show that if slippage with constant friction is assumed, the results do not differ greatly.

Take the simple case of a truck with two equal axles and assume that the motions are sinusoidal. In Fig. 5 consider the motion of the front wheel across the track, along the path A_1B_1. The motion is uniformly to the right, so that if the friction force is also to the right, the front wheel will gain energy, but if it is to the left the front wheel will lose energy. In the position shown, both wheels are losing energy and will continue to do so until they become tangent to their paths or their directions of motion

change. An examination will show that the lowest critical speed is obtained by letting the rear wheel touch the left rail as the front wheel crosses the center line of the track tangentially to its path. Then, when the front wheel passes C_1 its loss will become gain. Similarly, the rear wheel will start to gain energy after it passes A_2. Each wheel gains energy over half its lateral path and loses it over the other half so that the gain and loss are balanced. This condition for minimum critical speed is expressed by the formula

$$2l = V_{cr}/\omega \text{ or } V_{cr} = 2l\omega$$

which is the same formula as was obtained by the creepage theory for this case.

More complicated wheel-base arrangements can be calculated

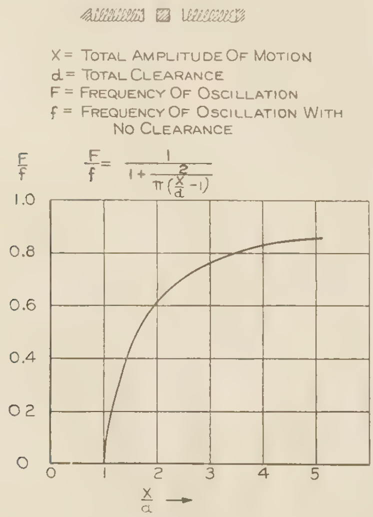

Fig. 6 Effect of Clearance and Amplitude on Frequency of Oscillation

analytically if required, using vector diagrams to portray phase relationships.

Frequency of Locomotive Oscillations

The frequency of roll is given by the well-known formula

$$f = (1/2\pi)\sqrt{[(M - Wh)g/Wk^2]}$$

where f = frequency, cycles per sec; M = the restoring moment, ft-lb per radian; W = the weight of the rolling part, lb; h = the height of the center of gravity of the rolling part above the center of rotation (center line of the axles), ft; k = the radius of gyration of the rolling part about a longitudinal axis through the center of rotation, ft; and g = acceleration due to gravity, 32 ft per sec per sec.

The ratio M/W is called the metacentric height h_m, and therefore the last given equation can be written

$$f = (1/2\pi)\sqrt{[(h_m - h)g/k^2]}$$

In some cases, the natural frequency of roll is also the frequency of nosing oscillations. This is true particularly of locomotives consisting of a single rigid frame, with a high center of gravity and a fairly low frequency of roll. When the flange of the rear driver of such a locomotive hits the rail, the rolling oscillation keeps this flange in contact until the return of the center of gravity allows the flange to leave the rail and the axle to move quickly across the track, where a roll in the opposite direction takes place.

If the frequency of roll is very high, the time taken by the roll will be short and the time taken by the axle to pass through the clearance will become important. To illustrate this, Fig. 6 shows the great effect which amplitude of motion has on frequency for the simplified case of a weight moving between springs, with clearance.

At the other extreme is a locomotive with an extremely low center of gravity, in which case the flange impacts do not produce any appreciable rolling and the frequency depends entirely on the speed with which the axles traverse the clearance. In the study of critical speed, it was shown that for a self-sustaining oscillation

$$V = Cf/\cos \epsilon$$

where V = speed, C = a constant depending on the type of locomotive, f = frequency of motion, and $(\pi/2) - \epsilon$ = phase difference between lateral and rotational motions. If $\cos \epsilon = 1$, then

$$f = (1/C)V$$

so that the frequency would increase with speed. The lateral blows on the flanges, which produce the necessary acceleration would be roughly proportional to the square of the speed. Actually, with extreme oscillation, the locomotive would tend to be thrown more nearly broadside onto the flanges, i.e., the lateral and rotational motions would come more nearly into phase. Then $\cos \epsilon$ would be less than unity and the frequency of sustained oscillations would no longer be proportional to speed. Nevertheless, the highest frequency oscillation which could be sustained is given by $f = (1/C)V$ and this will give an upper limit to the severity of the blows at any speed if rolling or other lateral motions in the locomotive are negligible. Detail calculations are necessary to proceed further. In starting the study of a locomotive, vector diagrams are very useful, as long as their limitations are recognized.

Another useful approximation is to consider the front end of the locomotive fixed against one rail and to calculate the period of swing of the wheel base as a damped pendulum, under the effect of the creepage forces. In some cases the motion can be divided into two types, to each of which an approximation applies.

Operation on Curves

Locomotives may oscillate at high speeds on curves of long radius. The energy methods of calculation are frequently useful but there are other ways of attacking the problem which are often more direct. The first stage in any study is to find the equilibrium position of the locomotive on the curve. For curves of very long radii the creepage theory is probably the most correct to use. For sharp curves, the theory of slipping is more correct. For preliminary studies it is generally very enlightening to take, for convenience, the theory of constant slipping force and to neglect the effect of the width of the gage. Then any wheel which slips is subject to a lateral force equal to the weight on the axle multiplied by the coefficient of friction. This coefficient may conventionally be taken as 25 per cent for slow speed on sharp curves (roughly up to 40 mph and on 6-deg curves and sharper), and 15 per cent for high speed on long-radius curves (roughly above 60 mph and up to 4-deg curves).

The locomotive wheel base is laid out in a possible position on the curve in question and the forces are evaluated. If the forces are in equilibrium the position has been correctly chosen. If the forces have a resultant, the locomotive is moved in the direction which this resultant indicates until finally an equilibrium position is located.[5]

When the equilibrium position is obtained it should be examined to see if it is stable. Stability is shown particularly by

two well-defined points at which the locomotive is held against the rail, these points being well separated to give a long base. If there is only one definite point and the locomotive is balanced by slipping wheels, guiding-truck restraints which are not locked, etc. the locomotive will be unstable at even slow speeds, because the frictional and other forces are not actually constant. Every slipping wheel will, if it can, roll and slip alternately, following a saw-tooth path. Therefore, the equilibrium position must be independent of large changes in the coefficient of friction. Also, wheels which make considerable angles with the rail should be able to execute small rapid oscillations without forcing the main mass of the locomotive to follow them, otherwise major oscillations will be started by small track irregularities.

A frequent cause of oscillation is due to excessive speed where the centrifugal force becomes sufficient to shift one or more axles from the inner to the outer rail, thus upsetting the balance and causing the locomotive to oscillate. More detailed calculations of any kind of motion which is indicated by this preliminary study can be made using energy methods as used for critical speed on tangent track.

Safety

Lateral pressure on an axle may (1) cause derailment, (2) spread or overturn the rail, (3) move the track bodily through the ballast, (4) break the axle, frame or other parts of the locomotive, or (5) may overturn the locomotive.

Derailment. The formula for the maximum lateral pressure on an axle which can be exerted before the wheel flange will climb the rail is,[6] for a positive angle of attack

$$H/W = [(\tan\alpha - f)/(1 + f\tan\alpha)] - (W'/W)f$$

For the flange tangent to the rail

$$H/W = \tan\alpha$$

The formula for the positive angle should be used in order to be safe. For a negative angle of attack

$$H/W = [(\tan\alpha + f)/(1 - f\tan\alpha)] + (W'/W)f$$

In these formulas H = the lateral force on the axle to produce derailment, W = the vertical load at the rail on the wheel which derails, W' = the vertical load at the rail on the other wheel of the same axle, α = the inclination to the horizontal of the plane of contact between the flange and the rail, and f = the coefficient of friction. The vertical loads are not the static loads but are those actually existing at the time of derailment.

The coefficient of friction is conventionally taken as 0.25. For high speed and small angles of attack it will probably be less although for slow speed on very sharp curves it may be more. It is suggested that for negative angles of attack, where decreased coefficient of friction decreases the allowable lateral load, 0.15 should be used for high speeds and small negative angles of attack.

The angle α, which is the inclination of the plane of contact between flange and rail, depends on the contour of the flange and of the rail, and on the angle of attack.

While the contour of the flange can be drawn, it is very laborious to determine graphically the effect of angularity between the plane of the wheel and the plane of the rail. The following semi-graphical method can be used where the angularity is enough to make it worth while.

The contour of the tire is drawn. Any convenient point on the contour is chosen as the origin of the coordinates and the contour is then a curve connecting x and y as shown in Fig. 7.

[6] "Traité de Stabilité du Matériel des Chemins de Fer," by Georges Marié, Ch. Béranger, Paris, France, 1924, pp. 288–299.

If the wheel is now turned through an angle θ about a vertical axis, the projected contour becomes a new curve, determined by X and Y as before and also by Z, perpendicular to the plane of projection.

Then, corresponding to any original point (x, y) there is a new point on the projected contour (X, Y, Z) given by

$$\begin{aligned} (R - Y)^2 &= (R - y)^2[1 - (y^1 \tan\theta)^2] \\ X &= \cos\theta[x + (R - y)y^1 \tan^2\theta] \\ Z &= -\sin\theta[x - (R - y)y^1] \end{aligned}$$

where R = the radius of the wheel measured to the point on the tire which is taken as the origin of the coordinates, and $y^1 = dy/dx$ in the original curve $y = f(x)$.

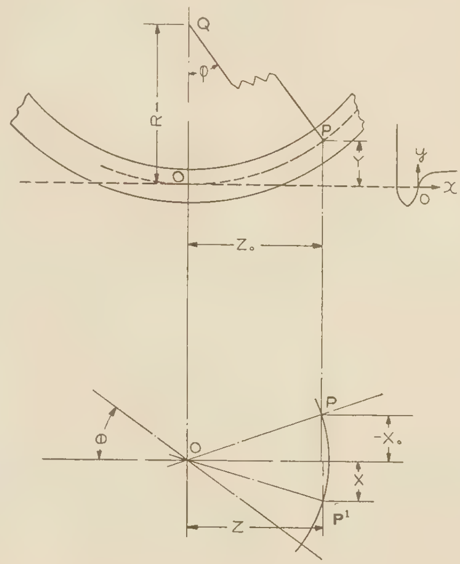

FIG. 7 DIAGRAM OF TIRE CONTOUR ON AN INCLINED WHEEL

Take a number of points (x, y) on the original contour, calculate X and Y by the latter three formulas, and plot Y against X. This will be the projected contour of the tire when turned through an angle θ. The projected contour can then be laid against a section of the rail to determine where they touch, and the slope of the tangent at the point of contact can be measured or calculated. If required, Z can also be calculated for this point to ascertain how far ahead of or behind the center of the wheel the point of flange contact is located. To prove the validity of these formulas refer to Fig. 7. Take the origin of coordinates at a point O on the tire. Let P be any point on the surface of the tire, with coordinates X_o, Y, Z_o. In the normal cross-section of the tire, P would have coordinates x, y.

Now rotate the wheel about a vertical axis OQ through an angle θ so that P moves to P^1. The coordinates of P^1 are X, Y, and Z. Then

$$X_o = x$$

$$QP^2 = (R - y)^2 = Z_o^2 + (R - Y)^2 \quad [1]$$

$$X_o^2 + Z_o^2 = X^2 + Z^2 \quad [2]$$

$$\tan\theta = \frac{(-X_o/Z_o) + (X/Z)}{1 + (X_oX/Z_oZ)} \quad [3]$$

A condition when X, Y, and Z are on the projected contour of the tire is expressed by

$$(\partial X/\partial Z)_{Y=\text{const}} = 0 \quad [4]$$

From Equations [1], [2], and [3]

$$(R-y)^2 - (R-Y)^2 = (Z\cos\theta + X\sin\theta)^2$$

and

$$x = X\cos\theta - Z\sin\theta$$

Differentiating with respect to Z, keeping Y constant and putting $(\partial X/\partial Z)_Y = 0$

$$-2(R-y)(\partial y/\partial Z)_Y = 2\cos\theta(Z\cos\theta + X\sin\theta)$$

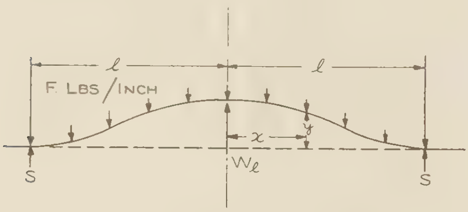

FIG. 8 TRACK DISPLACED BY A STATIONARY LOAD

and

$$(dx/dy)(\partial y/\partial Z)_Y = -\sin\theta$$

Eliminating $(\partial y/\partial Z)_Y$

$$(R-y)(dy/dx)\tan\theta = Z\cos\theta + X\sin\theta$$

Therefore

$$(R-Y)^2 = (R-y)^2[1-(y^1\tan\theta)^2] \qquad [5]$$

$$X = \cos\theta[x + (R-y)y^1\tan^2\theta] \qquad [6]$$

$$Z = -\sin\theta[x-(R-y)y^1] \qquad [7]$$

If the contour can be expressed by a formula, the projection can be calculated directly. For example, if the contour is conical $y = x\tan\alpha$ and the slope of the projected cone is

$$\tan\alpha/\cos\theta\sqrt{(1+\tan^2\alpha\tan^2\theta)}$$

MOVING TRACK

An excessive lateral load may move the track bodily through the ballast. The resistance to this motion is the friction between the ballast and the bottom of the ties.

A certain amount of side thrust can be exerted without any motion of the ties because the bending and twisting of the rail and compression of the ballast will distribute the thrust over a number of ties and the thrust on at least one tie must overcome the total friction on that tie. There is little data available as to the distribution of lateral pressure between ties up to the point where they slip. After slipping starts, the following simplified calculation is useful.

Assume the frictional force resisting side motion is constant and equal to F lb per in. Consider also, the case where deflection is all in one direction. In order for this to be possible, with a single applied load, reaction forces S are required at the points where deflection starts. Let the total length of deflected track be $2l$ in. Let the lateral load be W_l. Then $S = Fl - (W_l/2)$ as can be observed from Fig. 8.

For equilibrium and positive values of x, taking the origin at the point of application of the lateral load to the undeflected rail

$$y = \frac{9}{2048}\frac{W_l^4}{F^3EI_l}\left[1-6\left(\frac{x}{l}\right)^2+8\left(\frac{x}{l}\right)^3-3\left(\frac{x}{l}\right)^4\right]$$

where E = the modulus of elasticity of rail steel, I_l = lateral moment of inertia of the track (two rails), $l = 0.75(W_l/F)$, and y = the lateral deflection of the track.

The maximum deflection at the load is

$$y_o = 0.0044(W_l^4/F^3EI_l)$$

Field tests show that this equation represents the facts fairly accurately for deflections from 0.25 in. to 1 in. For small deflections, the loads are less than given by the equation. It appears that deflections up to approximately the order of 0.25 in. can be produced in heavily loaded track without causing permanent slip. In field tests, track was shifted considerably under a load and returned practically to the original position when the load was removed.

If the vertical load on an axle is W_v, the vertical force on the ballast per inch at a distance x is known to be

$$P = (\pi/8)(W_v/x_l)e^{-(\pi x/4x_l)}[\cos(\pi x/4x_l) + \sin(\pi x/4x_l)]$$

where $x_l = (\pi/4)\sqrt[4]{(4EI_v/U)}$, I_v = the vertical moment of inertia of one rail, and U = the modulus of foundation of one rail.

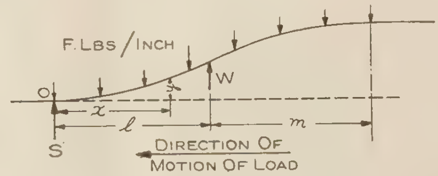

FIG. 9 TRACK DISPLACED BY A MOVING LOAD

The maximum pressure under load is

$$P_o = (\pi/8)(W_V/x_l)$$

Dividing the frictional force F by the vertical force P_o, an index of the nature of a coefficient of friction is obtained. Numerous tests on good track with wooden ties in stone ballast show values of F/P_o ranging from approximately 0.9 to 1.9 according to the condition of the track. The value increases with age up to the point where the ties decay if the track is undisturbed. If the track is raised and the ballast disturbed, the index is reduced. Observations on some metal ties indicate a materially lower resistance to lateral motion. Except on newly laid track or metal ties, it seems reasonable to rely on a ratio of at least 1.0.

Actually, the lateral friction force F will not be constant along the rail but will decrease as the vertical pressure on the ties decreases and will be affected by adjacent wheels. Except on the first wheel of a locomotive, however, the adjacent wheels tend to keep the track uniformly loaded and therefore the assumption of a constant force F per inch is reasonable. A leading wheel will move track a little more easily than these approximate formulas indicate. The previous discussion refers to static conditions. When the locomotive is moving, less lateral force is required to shift the track.

In Fig. 9, let O be the origin of the coordinates. Consider the track moving in one direction only and remaining shifted after the lateral load has passed. Let the coordinates move with the same speed as the load. Then, for the moving section

$$EI(d^2y/dx^2) = Sx + W\{x-l\} - (Fx^2/2)$$

where $\{\ldots\}$ are Macauley brackets, i.e., the bracket is taken as zero whenever the bracketed term is negative.

But

$$S = F(l+m) - W$$

Therefore

$$EI(d^2y/dx^2) = [F(l+m)-W]x + W\{x-l\} - (Fx^2/2)$$

and since $(dy/dx) = 0$ when $x = 0$

$$EI(dy/dx) = [F(l+m) - W](x^2/2) + (W/2)\{x - l\}^2 - (Fx^3/6)$$

Also putting $(dy/dx) = 0$ when $x = l + m$

$$(F/3)(l+m)^3 = (Wl/2)(l+2m)$$

and taking moments about the origin

$$Wl = (F/2)(l+m)^2$$

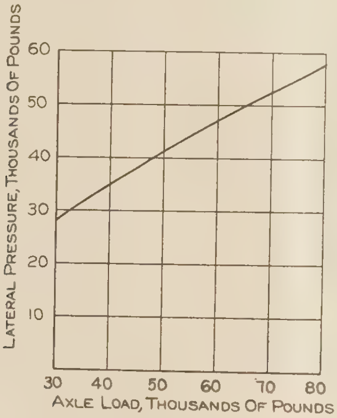

Fig. 10 Lateral Pressure From a Moving Load Required to Produce 0.25-In. Track Movement
($F/P_o = 1$; $W_l = 12.2W_v^{3/4}$.)

whence $l = 2m = (8/9)(W/F)$

Integrating again and putting $x = l + m$, the maximum deflection is found to be

$$y_{max} = (32/2187)(W^4/F^3) = 0.0146(W_l^4/F^3EI_l)$$

Otherwise, for the same maximum deflection, the ratio of moving force to static force will be

$$\sqrt[4]{(0.0044/0.0146)} = 0.74$$

The lateral force exerted by a moving locomotive axle to produce a 0.25-in. displacement of the track may be used as a convenient index. According to the previously given formulas

$$0.25 = (0.0146W_l^4/(F^3EI_l)$$

and

$$F = (F/P_o)(W_v/2)\sqrt[4]{(U/4EI_v)}$$

Therefore

$$W_l = (E^{1/16}/1.07)(F/P_o)^{\frac{3}{4}}(I_l^{\frac{1}{4}}/I_v^{3/16})U^{3/16}\,W_v^{\frac{3}{4}}$$

and if $E = 30{,}000{,}000$ lb per sq in. and $(F/P_o) = 1$

$$W_l = 2.74(I_l^{\frac{1}{4}}/I_v^{3/16})U^{3/16}\,W_v^{\frac{3}{4}}$$

The latter formula indicates that the weight of the rail has very little influence, that the modulus of foundation, i.e., the stiffness of the ballast, is quite important, and that, other things being equal, the allowable ratio of lateral to vertical load on an axle decreases as the axle load increases.

Taking typical values for good track with 131-lb rail, $I_l = 16.1 \times 2 = 32.2$, $I_v = 89$, and $U = 2500$. These values substituted in the last given equation give

$$W_l = 12.2W_v^{\frac{3}{4}}$$

the curve for which equation is plotted in Fig. 10.

In this discussion, the moment of inertia of the track I_l has been taken as being due to the rails alone, which is not strictly accurate inasmuch as other factors enter into it. Before these factors can be taken into consideration, additional experimental data is required, which is not available at the present time.

Lateral Oscillations of Rail Vehicles

By B. F. LANGER[1] and J. P. SHAMBERGER,[2] EAST PITTSBURGH, PA.

The authors point out in this paper that lateral oscillations, which either do not occur or are negligible at slow train speeds, are of vital importance at the high train speeds now demanded by the railroads in order for them to compete successfully with the swiftly moving automobile and the much swifter airplane. Aside from collisions and broken rails, practically all railroad accidents result from lateral derailments, since vertical derailments occur only on those rare occasions when a bridge or trestle collapses from flood or storm. Lateral derailments are caused by lateral pressure of wheel flanges against the rail. Therefore, the prevention of lateral derailments requires a knowledge of both the conditions which cause high lateral forces and of the conditions which must prevail in order to keep the lateral forces below some indicated safe value.

The literature on the subject of lateral oscillations of rail vehicles is rather meager. Nadal[3] and Marie[4] discuss the type of oscillations that are forced. Boedecker,[5] as well as the authors previously mentioned, discusses the weaving motion of the wheel base due to the conicality of the wheel treads. The weaving action which Boedecker discusses must be started by an initial disturbance and, in the absence of further periodic disturbances, can neither build up to a higher amplitude than that produced by the initial conditions nor be sustained, but must quickly die away. Carter[6] develops the stability condition for a rail vehicle as determined by the slippage at the treads.

The essential difference between the previous studies[3,4,5,6] and the present one is that the previous ones discussed forced oscillations the life of which depended upon the application of some periodic force, such as that from cylinder action or rail joints, whereas the present discussion describes and explains a type of oscillation which, after an initial disturbance, even though a minor one, may build up to dangerous proportions and sustain itself indefinitely on absolutely perfect track. This type of oscillation is frequently encountered and is commonly known as "nosing." It is only one phase of the whole problem, but it represents the most serious menace to the safe operation of rail vehicles at high speed.

1—FUNDAMENTAL CONSIDERATIONS

IN ORDER to investigate the possibilities of self-induced lateral oscillations in rail vehicles, the simple case of a symmetrical vehicle with two axles in a rigid frame supporting a spring-borne mass as shown in Fig. 1, will be considered first. It runs on a track the vertical flexibility of which is small compared to that of the springs of the vehicle, but which allows lateral motion of the wheels and axles, partly through flange clearance, partly through the riding-up of the wheel onto the throat of the flange, and partly through elastic distortion of the track itself. The springs of the vehicle allow vertical motion of the sprung mass relative to the unsprung mass. They also allow a sidewise rolling motion and a fore-and-aft pitching motion, but no relative rotation about a vertical axis. This system has six degrees of freedom, three of which are motions of sprung mass relative to unsprung mass, and three of which are motions of unsprung mass relative to the ground. The three motions of sprung mass relative to unsprung mass are (1) vertical motion, (2) rotational motion about a transverse axis which is defined as "pitching," and (3) rotational motion about a longitudinal axis which is defined as "rolling." (Since the height of this longitudinal axis above the rail is less than the height of the center of gravity, the roll results in a lateral displacement of the center of gravity of the sprung mass relative to the unsprung mass.) The three motions of the unsprung mass relative to the ground are (1) lateral motion, (2) longitudinal motion, and (3) rotational motion about a vertical axis. The system is not linear, the principal nonlinearity being in the lateral spring scale of the track.

As a first step in the investigation of the possibilities of self-induced oscillations, the system can be simplified considerably in order to determine which features of it are essential to the existence of the oscillations. Consider the case of a vehicle without any springs and no flange clearance. This eliminates the first three of the six degrees of freedom, which are the three motions of sprung mass relative to unsprung mass, and makes the lateral spring scale of the track linear. There are now just two possible types of lateral motion. One is a rotation about some vertical axis and the other is lateral translation of the wheel base. As far as lateral motions are concerned there are just two degrees of freedom. In order to have self-sustained oscillations in a system with two degrees of freedom there must be transfer of energy between the two degrees of freedom and also net energy input to the system. In other words, the following three conditions must all be satisfied: (1) The rotational motion must produce lateral translation; (2) the lateral translation must produce rotation; and (3) a periodic force must have a component in phase with the velocity of lateral translation and a periodic moment must have a component in phase with the velocity of rotation.

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[3] "Theorie de la Stabilite des Locomotives, Part 2; Mouvement de Lacet," by M. J. Nadal, Annales des Mines, vol. 10, 1896, pp. 232.

[4] "Traite de Stabilite du Materiel des Chemins de Fer," by Georges Marie, Librairie Polytechnique, Ch. Beranger, Paris, 1924.

[5] "Die Wirkungen zwischen Rad und Schiene," by Boedecker, Hahn'sche Buchhandlung, Hannover, 1887, p. 97.

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

[6] "On the Stability of Running of Locomotives," by F. W. Carter, Proceedings, Royal Society of London, vol. 121, 1928, p. 585.

As long as there is forward velocity and friction at the treads of the wheels, the first condition is satisfied. The vehicle tends to travel in the direction of its own center line rather than in the direction of the track center line, because of the friction at the treads. Thus, whenever there is an angular displacement between the track and vehicle center lines, the forward motion has a lateral component of which all points on the vehicle partake equally or, in other words, a lateral translation. Therefore condition (1) is satisfied.

In order to ascertain if the second condition is satisfied, see Fig. 2. Here the wheel base is in an angular position and the track is bowed out on one or both sides. There is a moment exerted by the track tending to restore the wheel base to its neutral position. Note, however, that this moment is a function of only the angular

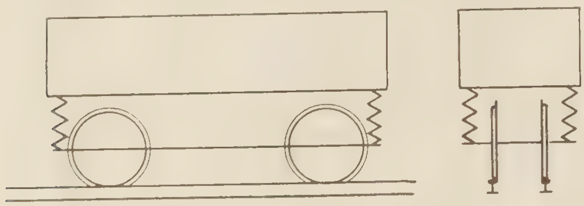

Fig. 1 Simple Rail Vehicle

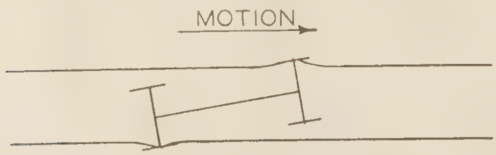

Fig. 2

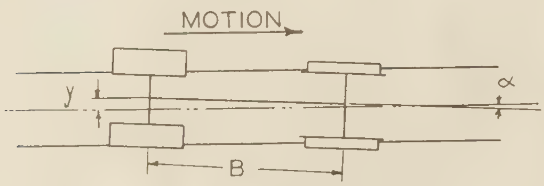

Fig. 3

position of the wheel base and lateral translation does not affect it. If the wheel base is translated laterally, say upward, in Fig. 2, the force at the leading axle increases but the force at the trailing axle decreases by the same amount and the moment remains the same. Inspection readily shows that this holds true for any position of the wheel base. Consequently, condition (2) is not satisfied. The lateral translation does not affect the angular motion and no self-induced oscillations are possible in this simplified system.

Now retrace one of these steps and replace the nonlinearity of the lateral resistance of the track. Actually, this nonlinearity is always present, except under very special conditions, because of the flange clearance. The spring scale may be considered as one which increases with displacement. Condition (1) is still satisfied. In Fig. 2, however, if the whole wheel base shifts upward, the leading axle moves in the direction of increasing spring scale, and the trailing axle moves in the direction of decreasing spring scale. Therefore, the force at the leading axle increases more than the force at the trailing axle decreases, and the moment is changed. This change in moment occurs for any positions of the wheel base except those in which its center line is parallel with that of the track.

Thus one condition is disclosed which is essential to the existence of self-induced lateral oscillations in rail vehicles. *The lateral resistance of the track must increase with the lateral displacement of the wheel from its neutral position. This condition is satisfied if flange clearance is present.*

It has been shown that for a conventional rail vehicle the nonlinearity of the track spring scale must be considered since it furnishes the coupling between lateral and rotational motion. The differential equations which exactly represent such a system are nonlinear and impossible to solve for the general case. As an approach to the problem it is interesting to consider a rigid two-axle rail vehicle with flanges on only the leading pair of wheels. Such a vehicle can sustain its oscillations on a linear track without clearance, and the equations which represent its motion are solvable.

In Fig. 3, let y = the lateral displacement between the center of the trailing axle and the center line of the track, α = the angular displacement between the track center line and the vehicle center line, V = the forward velocity of the vehicle, J = the polar moment of inertia of the vehicle about a vertical axis through the center of gravity, M = the mass of the vehicle, K = the lateral spring scale of the track, B = the wheel base of the vehicle, and c = the track damping constant.

In Fig. 3 let positive forces and displacements be upward and positive moments and angles counterclockwise. The lateral forces at the leading axle are $-K(y + B\alpha) - c(\dot{y} + B\dot{\alpha} - V\alpha)$. The lateral forces at the trailing axle are $-c(\dot{y} - V\alpha)$.

Note that the sliding velocity of a wheel is not its total lateral velocity, but the difference between the total lateral velocity and $V\alpha$. Longitudinal sliding has been disregarded.

The equilibrium of lateral forces gives

$$M[\ddot{y} + (B/2)\ddot{\alpha}] = -K(y + B\dot{\alpha}) - c(2\dot{y} + B\dot{\alpha} - 2V\alpha) \quad \text{..[1]}$$

The equilibrium of moments about the rear axle gives

$$\ddot{\alpha}[J + (MB^2/4)] + (MB/2)\ddot{y} = -BK(y + B\alpha) - cB(\dot{y} + B\dot{\alpha} - V\dot{\alpha}) \quad \text{......[2]}$$

These equations can be solved completely, but this is not necessary for our present purpose. By the application of Routh's criteria of stability[7] the conditions under which the motion is unstable can be determined. We find that for stability

$$V < \frac{B[2Jc^2B^2 + 2KJ^2 + (1/2)Mc^2B^4 + (1/8)M^2KB^4]}{4J^2c + 2cB^2JM + (1/4)B^4M^2c} \quad \text{..[3]}$$

If $J = (B^2M)/4$, the case of the mass concentrated at the axles, then

$$V < B[(c/M) + (K/4c)] \quad \text{...........[3a]}$$

If $J = 0$, the case of a light swivel truck, then

$$V < 2B[(c/M) + (K/4c)] \quad \text{...........[3b]}$$

It is interesting to note that this critical speed, below which oscillations will die out and above which they will increase indefinitely, is directly proportional to the length of the wheel base. It also increases with increasing track stiffness and decreases with increasing mass of the vehicle. Combining these last two statements we may say that the higher the natural frequency of the vehicle on the track the higher the critical speed. The term "critical speed" as here used does not mean the speed at which the oscillations are a maximum, but merely a border-line value above which oscillations build up and below which they die out.

2—APPROXIMATE SOLUTION

The next step will be to develop an approximate solution for a more conventional type of vehicle. Take a two-axle vehicle the mass of which is supported on springs and the unsprung mass of

[7] "Mechanical Vibrations," by J. P. Den Hartog, McGraw Hill Book Co., New York, 1934, chap. 7.

which is negligible. It has flanges on all four wheels and has lateral flange clearance in the track. The lateral spring scale of the track is high compared to the stiffness of the springs in the vehicle. The friction between the wheel and the rail is constant, not proportional to velocity.

The difficulty in solving the problem lies with the nonuniformity of the track resistance, which changes from zero to a large value when a flange hits the rail. It is known, however, that whatever the nature of the track resistance, it is such as to transform lateral motion of the wheel base into angular motion. Therefore, if the wheel base has a periodic lateral motion, it will also have a periodic angular motion. Let it be assumed that this angular motion can be expressed by the function $\alpha = A\cos\omega t$. The effect on the following solution if α is some other function of t is discussed in Appendix 3.

Fig. 4 shows six successive positions of the wheel base and the body. Wherever the rotation is in such a direction as to make a pair of wheels move to the same side of the track as that to which the vehicle is pointed, that pair of wheels may be sliding in either direction, or else rolling. If the rotation is in the opposite direction, the wheels must be sliding in the direction of rotation. Thus the direction of sliding of the rear wheels is indeterminate in I and IV and that of the front wheels is indeterminate in III and VI.

Let Y_1, Y_2 = the flange pressure on the leading and trailing axles, respectively, G_1, G_2 = the friction forces at the leading and trailing axles, d = the lateral displacement of the body relative to the wheel base due to roll, Q = the vertical load on a wheel, μ = the coefficient of friction between a wheel and the rail, and K = the lateral spring scale of the body on the wheel base.

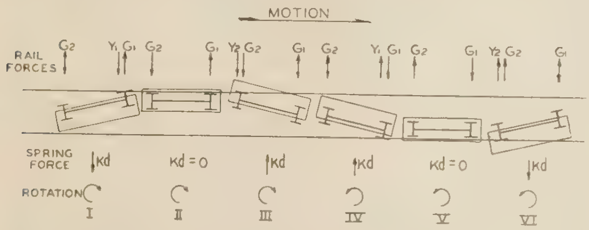

Fig. 4 Successive Positions of a Vehicle During Oscillation, Showing Direction of Rotation and Direction of Forces Exerted by the Track or Body on the Axles

The equilibrium equation for the forces on the wheel base is

$$Y_1 + Y_2 + G_1 + G_2 + Kd = 0 \qquad [4]$$

For condition I, $Y_2 = 0$, $G_1 = 2\mu Q$. Then $Y_1 + 2\mu Q + G_2 + Kd = 0$, or $Y_1 = -2\mu Q - G_2 - Kd$.

Since Y_1 must be negative, and d is at this time negative, and G_2 cannot exceed $2\mu Q$, then Kd must, in absolute magnitude, be equal to or less than $4\mu Q$. If the amplitude of roll is such that Kd exceeds this value, then both pairs of wheels will slide to the right and $Y_1 = 0$. At the instant when Kd decreases to the value $4\mu Q$, the leading flange hits the left rail and period I can start. Thus, the wheel base cannot turn until a definite point in the cycle of the body's oscillation is reached. Therefore, the rolling of the body on its springs acts as an escapement, controlling the frequency of the oscillation of the wheel base, and in the expression $\alpha = A\cos\omega t$, ω is 2π times the natural frequency of body roll.

The mass of the body will tend, on the average, to rotate about its center of gravity. During I and IV the friction forces will tend to throw the center of rotation toward the rear, and during III and VI toward the front, but during the whole cycle the net effect will be that of the center of rotation in the middle of the wheel base. If y is called the lateral displacement of the center of the wheel base, then the assumption can be made that $\dot{y} = V\alpha$. This assumption is based on the idea that the lateral translation, y, is produced by forward motion, V at a small angle α. $V\alpha$ is probably not the correct value for $\dot{y}$, but it represents an upper limit and gives the lowest possible value for critical speed.

If $$\dot{y} = V\alpha,\ y = \int_0^t VA\cos\omega t = (VA/\omega)\sin\omega t$$

Since y is the purely translatory motion of the point under the center of gravity of the body, it is y which excites the rolling oscillation of the body. When a damped system is forced to oscillate at its natural frequency, the vibration lags the disturbing force by 90 deg. Therefore, $d = -d_{max}\cos\omega t$, and the relative positions shown in Fig. 4 are correct.

Fig. 5

The question of whether or not the oscillation can sustain itself is dependent upon the relation between energy input and energy loss. Fig. 5 shows the vector relationships between the various motions and forces which have been considered. The sum of all the rail forces, Y_1, Y_2, G_1, and G_2 is equal and opposite to Kd. Therefore, the rail forces are in phase with $\dot{y}$, the velocity of lateral translation, and they constitute the input force.

The energy input per cycle of a periodic force in phase with a velocity is equal to π times maximum force times amplitude.[8] Therefore

$$\text{input per cycle} = \pi K d_{max} VA/\omega \qquad [5]$$

If, during any part of the oscillation, $Kd > 4\mu Q$ then during such time the motion y is not taking place and there is an interim during which there is no input. Therefore, $4\mu Q$ can be substituted for Kd_{max} in calculating the true input, provided $Kd_{max} > 4\mu Q$, and

$$\text{input per cycle} = 4\pi\mu QVA/\omega \qquad [5a]$$

The losses at the rail which drain energy from the oscillation consist of frictional losses due to sliding at the wheel treads. This sliding is due to (1) the angular oscillation about the center of the wheel base and (2) the lateral slippage of the whole wheel base when $Kd > 4\mu Q$. Considering (1) first, the force at each wheel is μQ and the distance through which it acts is $BA/2$ during each quarter cycle. Therefore

$$\text{loss per cycle} = 4(BA/2)\ 4\mu Q = 8BA\mu Q$$

The loss due to lateral slippage may be calculated by the curves of Appendix 1. In using these curves it must be remembered that the amount of slipping may be limited by the flange clearances.

Let us now study what happens for various amplitudes of roll, that is, various values of Kd_{max}.

If $Kd_{max} = 4\mu Q$

$$\text{input per cycle} = 4\pi\mu QVA/\omega \qquad [6]$$

$$\text{loss per cycle} = 8BA\mu Q \qquad [7]$$

For stability, $8BA\mu Q > 4\pi\mu QVA/\omega$, or

$$V < 2\omega B/\pi \qquad [8]$$

If $Kd_{max} < 4\mu Q$, the loss is the same as for $Kd_{max} = 4\mu Q$, but the input is smaller. Therefore, the critical speed is higher.

If $Kd_{max} > 4\mu Q$, the input is the same as for $Kd_{max} = 4\mu Q$,

[8] "Mechanical Vibrations," by J. P. Den Hartog, McGraw Hill Book Co., New York, 1934, p. 19.

but the loss is increased by the lateral slipping of the whole wheel base. Therefore, the critical speed is again higher.

This whole phenomenon may be illustrated by a curve such as that shown in Fig. 6, which was calculated for an actual locomotive. The curve ABC represents the critical speed for various initial amplitudes of roll. If the initial conditions for this locomotive are represented by either L_1 or L_2, the oscillation will die out. If they are represented by either L_3 or L_4, the conditions will change until the steady condition L_5 is reached, at which the input is equal to the loss.

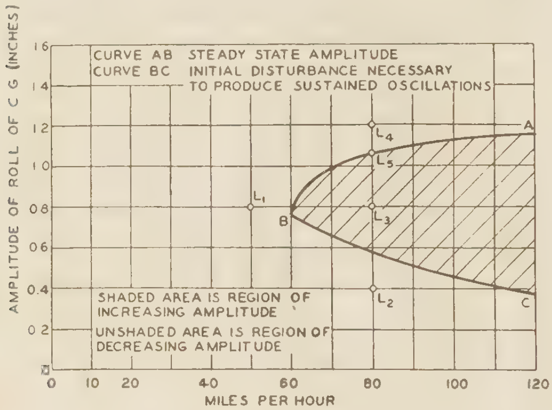

Fig. 6 Amplitude of Roll of an Atlantic-Type Locomotive

The most important critical speed is, of course, the lowest one, given in Equation [8]. Note that exactly the same conclusions regarding the effect of wheel base and natural frequency on critical speed can be drawn from Equation [8] as have already been drawn from Equations [3a] and [3b], which give the critical speed for the rigid vehicle with flanges on only the leading pair of wheels.

These same methods as were used can be applied to the case of a vehicle with more than two axles in a rigid wheel base. It is not necessary to follow the whole line of reasoning here. In Equation [9], the additional friction loss due to longitudinal sliding of the wheels is considered. The center of rotation should be taken at the center of the wheel base. Any other assumption would increase the friction loss and result in a higher critical speed.

If f = the natural frequency of the roll of the body, n = the number of axles in the rigid wheel base, x_i = the longitudinal distance from the center of rotation to an axle, s = the track gage, and the subscript i refers to the i-th axle in the wheel base, then

$$V_{cr} = [8f/n]\sum_{i=n}^{i=1}\sqrt{[x_i^2 + (s/2)^2]} \qquad [9]$$

Values of $\sqrt{[x_i^2 + (s/2)^2]}$ for various values of x_i are given in Table 1, Appendix 2.

In applying Equation [9] to a nonsymmetrical wheel base, the question of where to take the center of rotation arises. The safest assumption to make would be the one which gave the lowest value to $\Sigma\sqrt{[x_i^2 + (s/2)^2]}$. As far as the oscillations discussed in this paper are concerned, there seems to be no inherent advantage or disadvantage in the use of a nonsymmetrical as compared to a symmetrical axle arrangement.

The symbol f is defined as the natural frequency of the roll of the body. This is approximately the correct frequency to use for most simple vehicles of the rigid-frame type and can be calculated according to the equation

$$f = [1/2\pi]\sqrt{[(K_s b^2 - 2Wh)/2I_r]} \qquad [10]$$

where K_s = the spring scale for one side of the vehicle, neglecting cross-equalized springs, W = the sprung weight of the vehicle, I_r = the polar moment of inertia of the sprung weight about a longitudinal axis through the axle centers, h = the height of center of gravity of sprung mass above axle centers, and b = the lateral distance between springs.

Measurements have shown that the actual frequency at which the oscillation occurs is affected to a certain extent by the natural frequency of the locomotive in the track. The theoretical evaluation of this factor would require a much more detailed analysis than has been developed here; one which would take account of all possible degrees of freedom. It is highly advisable to use experimental data wherever possible in choosing a value of f for use in Equation [9].

Equation [9] was developed primarily for use on vehicles of the type of the rigid-frame locomotive. There is no reason why the same theory would not apply, however, to any individual rigid wheel base in a more complicated system, such as an articulated locomotive or a vehicle with swivel trucks. The only difference would be in the determination of the natural frequency which controls the oscillation. The authors' casual observations on street-railway cars with swivel trucks have led to the conclusion that the oscillations of the body are almost entirely angular about a vertical axis. Thus, the natural frequency would be determined not by the roll, but by the polar moment of inertia of the body about a vertical axis and the lateral restraint at the truck center pins.

If harmonic motion is assumed at all points of the vehicle, the center of rotation at a fixed point on the center line of the wheel base, and a known maximum lateral amplitude for a wheel, it is possible to solve the kinematics of the problem completely. This is done most easily by vector representation.

Let 2δ = the maximum possible full amplitude for a wheel (flange clearance plus rail deflection); A = the maximum value of α during a cycle; θ_i = the phase angle of i-th axle referred to center of rotation; and the other symbols the same as given previously.

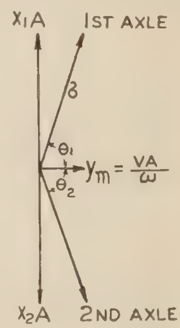

Fig. 7

Fig. 7 shows the vector diagram for a two-axle wheel base with the center of rotation in the middle. The value of y_{max} has already been determined. It is

$$y_{max} = VA/\omega \qquad [11]$$

The total amplitude of the i-th axle is seen to be

$$2VA/\omega \cos\theta_i \qquad [12]$$

Also, if the leading axle has the full amplitude 2δ,

$$\delta = VA/\omega \cos\theta_1 \qquad [13]$$

and hence

$$A = \delta\omega \cos\theta_1/V \qquad [14]$$

However

$$\cos\theta_1 = \frac{VA/\omega}{\sqrt{[(V^2A^2/\omega^2) + x_1^2A^2]}} = \sqrt{[V^2/(x_1^2\omega^2 + V^2)]} \qquad [15]$$

Substituting Equation [15] in Equation [14], we obtain

$$A = \delta\omega/\sqrt{(x_1^2\omega^2 + V^2)} \qquad [16]$$

It is also of interest to know the expressions for energy input

and energy loss in terms of the known constants of the vehicle. These are found to be:

$$\text{Input per cycle} = \pi k d_{\max} V\delta/\sqrt{(x_1{}^2\omega^2 + V^2)} \ldots\ldots [17]$$

or, at the critical speed, when $Kd_{\max} = 2n\mu Q$

$$\text{input per cycle} = 2\pi n\mu QV\delta/\sqrt{(x_1{}^2\omega^2 + V^2)} \ldots . [18]$$

and due to rotation the

$$\text{loss per cycle} = \frac{8\mu Q\delta\omega}{\sqrt{(x_1{}^2\omega^2 + V^2)}} \sum_{i=n}^{i=1} \sqrt{[x_i{}^2 + (s/2)^2]} \,.. [19]$$

3—DESIGN REQUIREMENTS FOR A HIGH-SPEED LOCOMOTIVE

The basic design requirement for a high-speed locomotive is that its critical speed be considerably above the highest speed at which the locomotive is expected to operate. The critical speed as given by Equation [9] is in error on the side of safety in that it assumes $\dot{y} = V\alpha$ and does not consider the energy loss from the friction in the equalization springs and at other rubbing surfaces that tend to damp the oscillation. The experience of the authors has been that the factor of safety introduced by these simplifying assumptions is sufficient for good practice, so that a locomotive can operate safely up to its calculated critical speed. The design features that influence steady performance at high speed are indicated in Equation [9], which is the equation for critical speed. For a given track gage s and a given number of axles n in the rigid wheel base, it is observed from Equation [9] that the inherent critical speed is a function of (1) the frequency of roll, and (2) the length of the rigid wheel base. A high frequency of roll and a long rigid wheel base produce a high critical speed. The critical speed of a locomotive may be raised or lowered to the extent that the designer can vary (1) the frequency of roll, and (2) the length of the rigid wheel base, these two factors being the ones which dominate the critical speed.

Equalization System

The equalization system is an important design feature as regards critical speed because of its influence on the frequency of roll. The equalization system influences the frequency of roll by (1) the scale of the spring, (2) the lateral distance between spring supports, and (3) the ratio between cross-equalization and side-equalization. The relative influence of these three factors on frequency is indicated by Equation [10] for natural frequency of roll. Other things remaining constant, the critical speed is raised by (1) increasing the stiffness of the springs, (2) by increasing the lateral distance between spring supports, and (3) by increasing the amount of side-equalization.

Stiff springs are desirable from a consideration of critical speed but are undesirable from a consideration of resulting forces on vehicle and track. Springs are employed in the equalization system in order to shield both the sprung weight and the track from forces in excess of those existing under static conditions. From the consideration of reducing forces on sprung weight and track, the softer the supporting springs the more effective they become. To stiffen the springs defeats the purpose for which they are used. The fact is that the designer can vary the stiffness of the equalization springs only within rather well-defined limits. To stiffen them defeats their purpose, to soften them beyond a certain limit introduces complications on account of the large vertical movement in the pedestal guides. It would seem, then, that the logical procedure to follow is to employ springs as soft as can be used from practical considerations without too great a sacrifice of frequency. From Equation [10], it is observed that the frequency of roll varies directly as the square root of spring stiffness. Because of this relationship between frequency and spring stiffness, it is possible to soften the springs of a locomotive considerably without producing more than a minor change in frequency.

The type of spring employed in the equalization system, that is, whether leaf-type or coil-type, is as important as the spring scale. Considerable frictional damping is introduced by leaf springs which is absent when coil springs are used. A satisfactory arrangement results from a combination of low-frequency leaf springs in series with high-frequency coil springs. In this arrangement the low-frequency leaf springs practically determine the period of roll and provide frictional damping while the high-frequency coil springs are placed directly over the journal boxes and absorb the high-frequency shocks from the road bed.

The lateral distance between supporting springs is fixed by the location of journal boxes. For a given angularity of roll, the greater the lateral distance between springs, the greater will be the variation in vertical load under the drivers, but as regards critical speed, the greater the lateral distance between springs, the higher will be the frequency of roll and the higher the critical speed. Other things remaining constant, outside journal boxes will produce a higher critical speed than will inside boxes. For steam locomotives, however, outside boxes are possible only with inside cranks, and in this country inside cranks are unknown except for three-cylinder locomotives. In Europe, inside cranks for steam locomotives are common. For electric locomotives, both in this country and elsewhere, outside boxes predominate.

With respect to the ratio between the amounts of cross-equalization and side-equalization, it is probable that the need for cross-equalization has been somewhat overemphasized in the past. The object of cross-equalization is to prevent a wide difference in the vertical loads under the two wheels of a given axle. But field tests have shown that cross-equalization of itself is unable to achieve this objective. Field tests indicate that for a locomotive with the customary amount of cross-equalization the vertical loads under the two wheels on the same axle vary the greatest under the condition of nosing. The prevention of nosing, then, should accomplish more toward equalizing vertical loads than does cross-equalization, and nosing is prevented not by cross-equalization but by side-equalization which, by influencing the frequency of roll, raises the critical speed.

From the standpoint, then, of critical speed and rail loads, the equalization system (1) should employ relatively soft leaf springs in series with stiff coil springs directly over the boxes, (2) the lateral distance between springs should be made large by placing the journal boxes outside the drivers if possible, and (3) the amount of side-equalization should be a maximum, and, if cross-equalization is insisted upon, restrict its amount to 25 per cent or less of the sprung weight.

Weight of Spring-Borne Parts

To determine the influence on critical speed, the absolute weight of the spring-borne parts must be considered in conjunction with the height of the center of gravity above the axle and the stiffness of the equalization springs. The weight of the spring-borne parts influences critical speed by influencing the frequency of roll. However, for a given center of gravity, the weight of the spring-borne parts may be varied at will without changing the critical speed provided the stiffness of the equalization springs is also varied. The spring-borne parts may thus be said to exert a neutral influence so far as critical speed is concerned. Furthermore, the weight of spring-borne parts is one of the design features that can be varied only within rather narrow limits when once the general characteristics of the locomotive have been determined. Hence, it becomes apparent that a varia-

tion of the sprung weight may as well be left out of consideration in the search for means of raising the critical speed.

Location of Center of Gravity of Spring-Borne Parts

The location of the center of gravity of the sprung mass is a most important factor in determining the critical speed, but like the absolute weight of the sprung mass, it is a design feature which can be varied only within exceedingly small limits for a locomotive with prescribed characteristics. Any consideration of the location of the center of gravity of the sprung mass is thus nonessential in the search for means for raising the critical speed. The higher the center of gravity of the spring-borne parts, the lower becomes the frequency of roll and the lower the critical speed. But, what can be done about it?

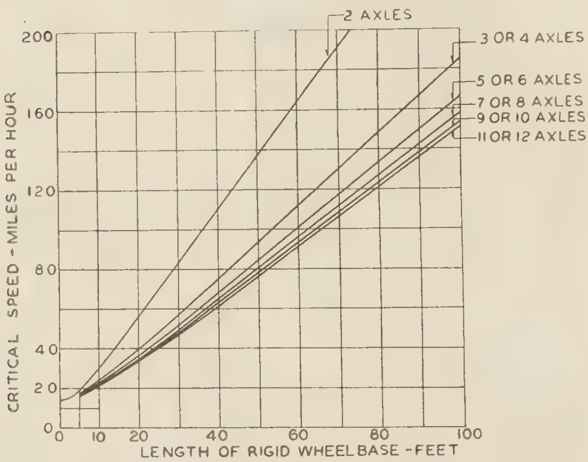

Fig. 8 Effect of the Length of a Rigid Wheel Base on Critical Speed for Various Numbers of Equally Spaced Axles Calculated From Equation [9], $f = 1.0$

The center of gravity of the sprung parts of an electric locomotive is usually somewhat lower than that for a steam locomotive. This lower center of gravity for the electric locomotive is frequently considered a disadvantage, but an analysis of the influence of the center of gravity on locomotive performance does not justify this belief. To determine the tendency from raising or lowering the center of gravity, let us assume extreme conditions. Assume first that the center of gravity is very high above the rail. For this condition the frequency of roll would be very low with a corresponding low critical speed, also a very small amount of roll would throw all of the weight on one rail and bring the center of gravity outside the gage, thus causing the locomotive to tip over. Now, assume that the center of gravity is on line with the center line of the axle. For this condition a lateral force would produce an unbalance of vertical loads at the rail, but this motion of the sprung mass would be purely rotational with no roll at all at the center of gravity. Hence, there would be no lateral translation of the center of gravity and, therefore, for this condition, lateral oscillations would be impossible because lateral translation of the sprung mass is absolutely necessary if the lateral oscillations are to be sustained. Finally, assume that the center of gravity is on line with the rail. For this condition, there is lateral rolling and lateral oscillations are possible but lateral forces now produce no change in the vertical wheel loads at the rail. Thus, the center of gravity of the sprung mass, from a consideration of lateral oscillations, should be located on the center line of the axle, but from a consideration of the unbalance of vertical loads at the rail, the center of gravity should be on line with the rail; that is, it should be in the plane in which the lateral force reacts on the locomotive. The conclusion is reached then that lateral oscillations would be impossible and there would be no unbalance in rail loads from lateral forces if the center of gravity of the sprung mass and the reaction on the locomotive from lateral forces both occurred at the center line of the axle.

For the conventional locomotive, the location of the center of gravity of the sprung mass is practically beyond the control of the designer. However, if possible to do so, move the center of gravity downward, not upward. Lowering the center of gravity raises the critical speed and decreases the amount of vertical-load unbalance at the rail. The lower center of gravity of the electric locomotive, as compared to the steam locomotive, is a distinct advantage.

Wheel Base

The length of the rigid wheel base is the most important design feature influencing critical speed. It has been stated by various writers in the past that a high-speed locomotive should have a long rigid wheel base, but the reason for this requirement has hitherto been unknown. This reason is revealed by an inspection of Equations [18] and [19] which are the equations for energy input and frictional losses during oscillations. These equations show that lengthening the rigid wheel base decreases the available energy input for sustaining oscillations and at the same time increases the frictional losses for damping them out. Hence, a long rigid wheel base becomes the principal requisite for every high-speed locomotive. The effect of increasing the length of the rigid wheel base is illustrated in Fig. 8.

Guiding Trucks

With the exception of the permanently rigid wheel base, the guiding trucks become the most important design feature determining critical speed. The guiding trucks determine critical speed because of their influence on the length of the rigid wheel base. The importance of guiding-truck design is best illustrated by considering two extreme values for the force necessary to produced initial lateral displacement of the truck, commonly known as truck crack-off.

Assume, first, that the crack-off is 10 per cent of the bolster load. For a truck with the usual amount of weight below the center pin, this would represent a lateral force at the rail of about 8.75 per cent of the vertical rail load. For truck wheels, with no tractive effort, the coefficient of friction against lateral slippage at the rail is considerably greater than 8.75 per cent for all normal operating conditions. This means it is possible to swing the truck bolster laterally without sliding the truck wheels. If the locomotive starts to oscillate laterally, the truck bolster will swing laterally within the truck frame without offering any damping force against the oscillations unless the guiding characteristic is so steep that the rail coefficient of friction against lateral slippage is exceeded during part of the oscillation cycle. The reason the lateral swing of the bolster offers no damping resistance is because the energy loss of the sprung weight in overcoming the truck resistance from central position to the extreme swing is immediately restored on the return swing so that the net effect is zero.

Assume now that the truck crack-off is 40 per cent of the bolster load. This represents a lateral force at the rail of about $35^1/_2$ per cent of the vertical rail load. A lateral force of this magnitude at the rail should be capable of sliding the truck wheels laterally across the head of the rail against the highest coefficient of friction that might be expected with the locomotive at speed. Experiments indicate that the coefficient of friction against the lateral sliding of a wheel in motion decreases with speed. Hence, a lateral force at the rail equal to $35^1/_2$ per cent of the vertical rail load should cause the wheels to slide laterally even at quite a

low speed. If the locomotive oscillates laterally, the truck wheels are slipped laterally across the rail head thus introducing an enormous damping force. So far as lateral oscillations are concerned, it is thus seen that a guiding truck with a crack-off high enough to slide the truck wheels laterally across the rail has the effect of lengthening the rigid wheel base to the truck center pin.

It is customary to design electric locomotives for two-way operation. Practically, though not necessarily, the specification for two-way operation demands a symmetrical-type locomotive. It is usual, then, for electric locomotives to have identical guiding trucks at both ends. From the standpoint of providing a turning moment on curves, this arrangement is more effective than might appear at first sight. The location of the center of rotation of a locomotive rigid wheel base is always such as to make the lever arm of the leading truck, with respect to the driver against the inner rail, longer than the corresponding lever arm of the trailing truck. This difference in lever arms produces a net guiding moment even for the condition in which both guiding trucks have the same amount of guiding, which is the case for trucks of the constant-resistance type. Therefore, it is evident that for electric locomotives it is possible to extend the rigid wheel base from truck center pin to truck center pin by employing trucks at both ends with a crack-off, with respect to the rail, greater than the lateral coefficient of friction without sacrificing adequate guiding for curves. If this initial crack-off resistance is too high for operation on sharp curves it is possible to give the truck restraint a drooping characteristic.

It is thus seen that a high-restraint truck possesses very great value as a means of extending the rigid wheel base, but this advantage of high restraint has been shown by experience to be virtually lost if the truck is free to swivel about its center pin, as is usually the case with four-wheel engine trucks. When free to swivel, the truck itself goes through an oscillation similar to that of the main wheel base and consequently runs back and forth through the track clearance without much sliding friction. This freedom of a four-wheel truck to run back and forth without sliding can be prevented by the application of an antiswiveling device such as a radius bar. To limit the flange pressures on extremely sharp curves and turnouts, the restraint of the radius bar on a four-wheel truck should be limited so that it becomes possible for the truck to swivel when its swiveling torque reaches some selected value.

In Equation [9] it is assumed that all axles have the same loading. If this is not the case, then the term $\sqrt{[x_i^2 + (s/2)^2]}$ for each axle must be given a weighting factor equal to the ratio between its actual load and the average load for all the axles being considered, and the symbol n instead of representing the actual number of axles represents the sum of the weighting factors. Letting k represent the weighting factor, Equation [9] may be written

$$V_{cr} = 8f\left[\frac{k_1\sqrt{[x_1^2 + (s/2)^2]} + k_2\sqrt{[x_2^2 + (s/2)^2]} + \ldots\ldots}{k_1 + k_2 + \ldots\ldots\ldots}\right] \qquad [20]$$

Since the expression $\sqrt{[x_i^2 + (s/2)^2]}$ increases in value with the increase in distance from the center of rotation, it follows that the heavier the axles at the ends of the locomotive, the higher becomes the critical speed. Since the end axles on a locomotive are usually truck axles, this means that the trucks should carry as much of the total load as is possible, considering wheel-load limitations and the weight requirements of the drivers for adhesion. The influence of changing the ratio between driver load and truck load is illustrated by curves A and B of Fig. 9. Another method for increasing the equivalent rigid wheel base of a locomotive is to give unrestrained lateral play to the intermediate axles of a rigid wheel base.[9] Unrestrained lateral play eliminates both the input and the loss from these axles. Since the input from an axle is fixed regardless of its position in the wheel base, and its loss is proportional to its distance from the center of rotation, this arrangement produces a considerable increase in the critical speed. The magnitude of the increase in critical speed, for a particular locomotive, by the use of unrestrained lateral is illustrated by a comparison between curves A and B of Fig. 9. In this figure, curve B represents the critical speed for a locomotive having the wheel arrangement as shown and for the condition of no free lateral, while curve A represents the critical speed for the condition of unrestrained lateral in driving axles No. 2 and No. 3.

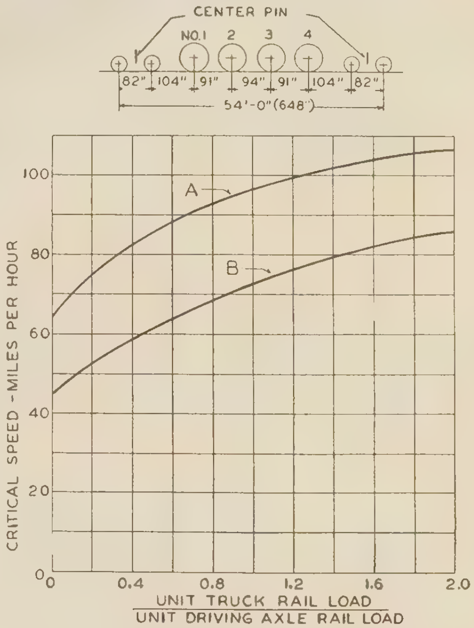

FIG. 9 EFFECT OF WEIGHT DISTRIBUTION ON CRITICAL SPEED CALCULATED FROM EQUATION [20], $f = 1.0$
(Curve A—Unrestrained lateral play at second and third drivers, Curve B—Normal minimum lateral play at all axles.)

RAIL LOAD

The absolute value of the rail load is not a factor in determining critical speed. The axle loading influences critical speed only to the extent that the weight of the sprung mass influences the frequency of roll.

GYROSCOPIC ACTION

The revolving wheels and axles of a locomotive give rise to gyroscopic forces whenever their axes are subjected to angular rotation. On an electric locomotive, the motor armatures constitute additional rotating masses to those of the wheels and axles.

A rotating mass which is part of the deadweight does not have any gyroscopic effect whatever on the oscillations described in this report. The rotation of the wheel base α may cause a shifting of load from one side to the other, but there can be no rolling oscillation of the deadweight, so the shifting of load does not result in an oscillation.

Spring-borne motors in an electric locomotive, however, can affect the oscillation because the rotation α of the wheel base causes the motors to exert gyroscopic forces which aid the rolling of the cab. Calculations show the magnitude of this effect to be small.

[9] Suggested to the authors by B. S. Cain.

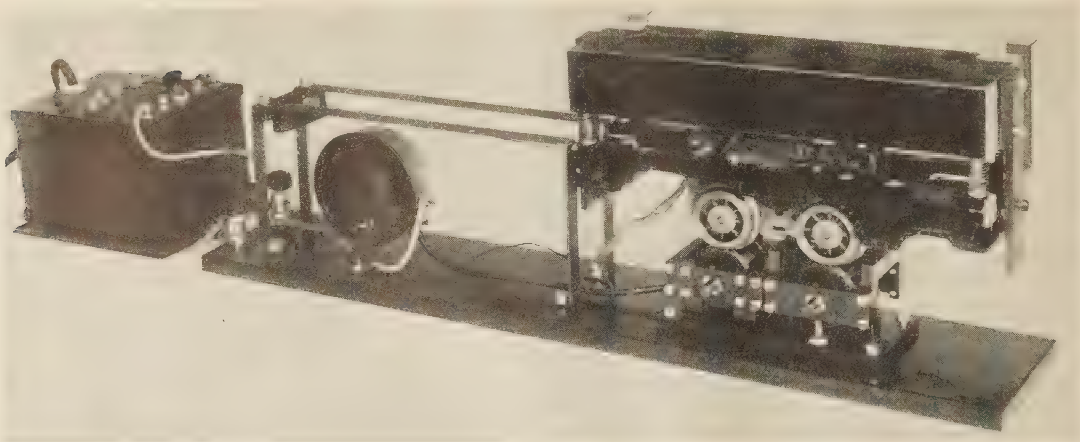

Fig. 10 Model for Illustrating Lateral Oscillations

Rail and Tire Contour

There has been discussion in the past concerning the effect of conical wheel treads on locomotive and car performance. In this paper, the coning of the wheel tread is not considered and the wheel is treated as having a tread that is a true cylinder. No matter what the contour may be on leaving the shop, under service conditions, the tread surface quickly conforms to the contour of the rail head.

Track Clearance

It is very important to keep the track clearance as small as possible. Although this quantity does not appear directly in the formula for critical speed, it does appear in the expressions for energy input and energy loss, both of which are directly proportional to track clearance δ. Therefore, at speeds above the critical, the larger the clearance the larger the amount of energy available to sustain and increase the oscillation. Thus, when the clearance is large there is less possibility of the spring friction being sufficient to control the oscillation.

Probably the most serious result of large track clearance is not shown by the equations, that is, its effect on the magnitude of the rail forces. The amplitude of the leading axle is equal to the track clearance. In any oscillation, velocity and acceleration increase proportionately to amplitude, and therefore the flange pressure increases in proportion to the track clearance, since it is the flange pressure that produces the lateral acceleration.

Therefore the smaller the track clearance the less the possibility of sustained oscillations and the less the damage they can do to the track if they do start. For the condition of zero track clearance, oscillations of the type discussed in this paper become impossible.

Coefficient of Friction

The effect of a low coefficient of friction is twofold. First, it lowers both the input and the loss energies and thus decreases the energy available to sustain oscillations. At the same time, however, it decreases the disturbance necessary to start sustained oscillation and thus makes the vehicle less stable. A low coefficient of friction against lateral slippage may be produced by wet rails or high tractive effort.[10] A very high coefficient of friction may be found in rubber-tired rail vehicles, which have recently been tried out.

[10] "Guiding and Running of Locomotive Wheels on Track," by J. Buchli, *Schweizerische Bauzeitung*, vol. 82, no. 10. September, 1923, pp. 119–125.

4—MODEL TESTS

In order to bring experimental evidence to bear upon the conclusions drawn from the theory without going at once to the expense of making tests on actual locomotives, a small model was constructed.

The critical speed of a model rail vehicle, according to Equation [9], is proportional to the natural frequency of roll of the sprung weight and the linear dimensions of the model. Therefore, if the natural frequency were made the same as that of a full-scale locomotive, the critical speed would be reduced in the same ratio as the scale of the model, and oscillations would occur at easily obtainable speeds.

An illustration of the model used is shown in Fig. 10. The first trials showed that it was not practical to run the model on a track. The track could not easily be made smooth enough nor the straight stretches long enough. Also, the forward motion of the model along the track made it difficult to study the lateral oscillations in detail. It was found that a much better arrangement was obtained by mounting the model on rollers. The model was restrained from longitudinal motion by a wire similar to a drawbar.

The model contained all the parts which, according to the theory, were necessary for the existence of lateral oscillations. It had two axles, both driven through gears by a single motor, and both mounted in the same rigid frame. It had a sprung weight supported from the frame by four helical springs, one at each corner. The center of roll of the sprung weight was made to coincide with the height of the axles by means of a pin and clevis arrangement at each end. The model had a wheel gage of $1^1/_4$ in., a wheel base of $2^5/_{16}$ in., a total length of 9 in., an unsprung weight of 2.6 lb, a sprung weight of 9.5 lb, a natural period of roll of 0.6 sec, and a critical speed of 1.9 fps.

The wheels had flanges and cylindrical treads. The rollers consisted of two cylinders, one under each axle. Each cylinder had two beads spaced $1^1/_4$ in. center to center for the wheel treads to run on and the flanges to bear against. These beads represented the rails. In order to equalize the weight on the four wheels, one roller was mounted on trunions.

The speed of rotation of the wheels, which represented the forward velocity of the model, could be varied by changing the voltage supplied to the motor. By the use of a rheostat any speed up to about 7 fps could be obtained.

Observed Oscillations

At speeds above 3 fps a rolling oscillation started by hand be-

came self-sustaining and was accompanied by lateral oscillations of the running gear at the same frequency. At speeds below 3 fps the oscillation would die out. It died out quickly at very low speeds but more and more gradually as the speed was increased. Therefore, 3 fps was the actual critical speed. This was higher than the calculated value of 1.9 fps, but this discrepancy was to be expected due to the various frictional losses in the mechanism not considered in the theory.

It was observed that the oscillation of the model was of the same nature as that predicted by the theory, but no exact comparison could be made by mere visual observation. In order to study the motion quantitatively and in detail, a motion-picture camera was used.

There were four points where the motion had to be measured. They were at the front and the back of the sprung and the unsprung weights. The camera was placed so as to look directly down on the model. The four points in which the authors were interested were brought into the same focal plane by bringing up extensions from each end of the unsprung frame to the level of the top of the sprung weight. Spots of white paint located the four points to be observed. Pictures were taken at the rate of about 30 per second, which gave a sufficient number of readings during the 0.6 second period of the oscillation to enable the authors to plot the curves of the motion. The time scale was determined by placing a stop watch in the picture alongside the model. Fig. 11 shows a portion of a strip of pictures from which readings were taken.

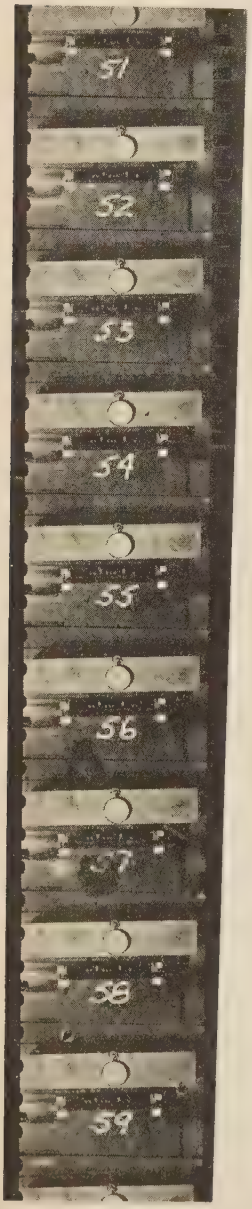

Fig. 11 Section of Motion-Picture Film of Model During Oscillation

The motions observed were (*a*) lateral motion of the front of the unsprung weight, (*b*) lateral motion of the rear of the unsprung weight, (*c*) lateral motion of the front of the sprung weight at the top of the model, and (*d*) lateral motion of the rear of the sprung weight at the top of the model. By means of these directly measured values, it was possible to derive (*e*) lateral motion of the top of the model due to roll by subtracting (*a*) from (*c*) or (*b*) from (*d*); (*f*) angularity of the wheel base α by subtracting (*b*) from (*a*) and dividing the result by the length of the model; (*g*) lateral motion of the center of rotation, which was approximately $^1/_2[(a) + (b)]$.

The primary motions are (*a*), (*b*), (*e*), (*f*), and (*g*). These five motions are plotted for one representative cycle in Fig. 12. It is interesting to note how well the experimental results check with the theory. The theoretical values of motions (*a*), (*b*), (*f*), and (*g*) are shown in Fig. 12 along with the experimental results. Motion (*e*), the roll, was not correctly represented by the model as far as its amplitude was concerned because the theoretical value greatly exceeded the solid position of the springs. The other motions, however, show a remarkably good check between theory and experiment. Note that the lateral slippage of the wheel base occurring at the instant of maximum roll shows up quite plainly in the experimental curves. This slippage was not considered in calculating the theoretical curves.

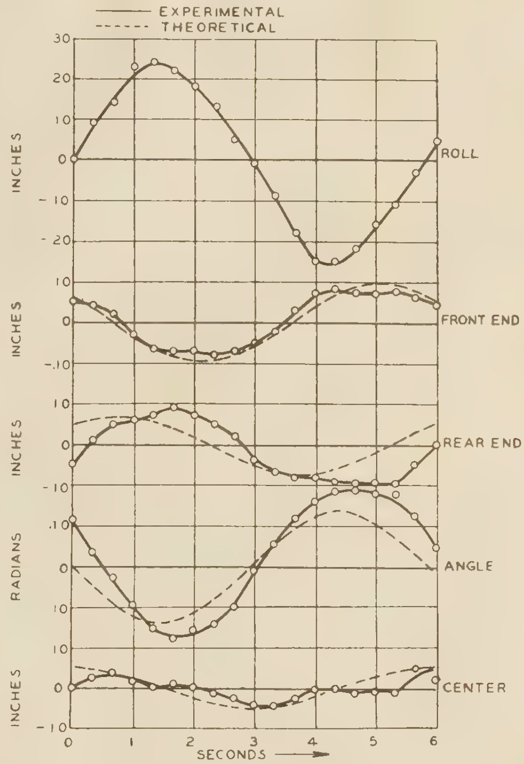

Fig. 12 Lateral Oscillations of Locomotive Model

Other tests on the model showed that the critical speed was proportional to the frequency of the oscillation, as predicted by the theory.

5—CONCLUSIONS

In this paper the authors have offered a mathematical theory which explains the self-sustained lateral oscillations of rail vehicles. The formulas derived from this mathematical analysis check with all available experimental data.

The conclusions to which the theory leads may be summarized as follows:

1 For satisfactory operation at high speed, a rail vehicle should have a critical speed, as calculated from Equation [9], no lower than its highest operating speed.

2 Recommended tendencies in design are: (*a*) as high a natural frequency of roll as is possible without sacrificing adequate vertical resiliency; (*b*) a long rigid wheel base; (*c*) high-resistance guiding trucks with limited swivel restraint. This applies to four-wheel trucks; two-wheel trucks are inherently nonswiveling; (*d*) loads concentrated as much as feasible on the end axles; (*e*) unrestrained lateral motion in axles near the center of the wheel base.

3 Recommended tendencies in track maintenance include:

(*a*) the elimination of irregularities. (*b*) The narrowing of the gage to decrease the clearance between flange and rail.

There are three factors to be considered in determining whether or not the performance of a vehicle will be satisfactory in a given service. These factors are (1) the inherent critical speed of the vehicle, (2) the maximum speed at which it will operate, and (3) the condition of the track. For the most satisfactory operation, the calculated critical speed should be well above the operating range, say 30 per cent above the maximum speed. The vehicle should then operate safely on ordinary track. For track that is better than ordinary, the ratio between critical and operating speed may be reduced. On perfect track, any vehicle can operate at any speed. However, it is not safe to rely entirely on a good track condition to maintain satisfactory operation, since the best of track may develop an occasional rough spot.

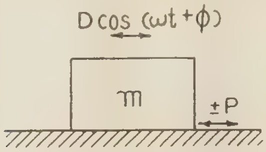

Fig. 13

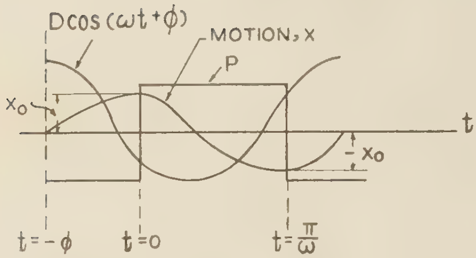

Fig. 14

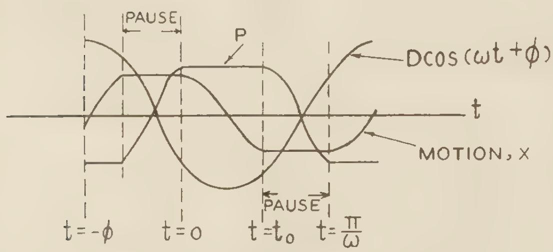

Fig. 15

The design of railway vehicles becomes of increasing importance with every increase in the speed of railway trains because the higher the train speed the greater becomes the effect on vehicle performance from track irregularities. However, in so far as self-sustained oscillations are concerned, it is believed that with proper vehicle design and with good track maintenance, it will be possible, with safety, to appreciably increase train speed above the general level now prevailing.

Appendix 1

Lateral Sliding of Wheels Due to Rolling of Body

The amount of lateral sliding of the wheel base which can be produced by the rolling of the cab may be calculated if the problem is simplified to that shown in Fig. 13. Here a mass m representing the total mass of the vehicle, is resting on a surface which resists any horizontal motion of the mass with a force P. A reciprocating force D cos $(\omega t + \phi)$, representing the inertia force of the body, is applied to the mass. What is the motion of the mass m for various values of the ratio D/P?

Obviously, if $D/P < 1$ there is no motion at all. If D/P is slightly greater than unity there will be an intermittent motion with a pause during each half cycle. If D/P becomes quite large, the mass can be conceived as having a continuous reciprocating motion with no pauses.

First consider the case where the motion is continuous, Fig. 14. If x is the displacement of the mass m from its mean position, the equilibrium conditions for the mass during its motion in one direction can be expressed as

$$m(d^2x/dt^2) - P = D \cos (\omega t + \phi)$$

$$\text{or, } d^2x/dt^2 = P/m + (D/m)\cos (\omega t + \phi) \quad [a]$$

Integrating

$$dx/dt = Ft/m + (D/m\omega) \sin (\omega t + \phi) + C' \quad [b]$$

Integrating again

$$x = Ft^2/2m - (D/m\omega^2) \cos (\omega t + \phi) + C't + C'' \quad [c]$$

The following conditions are known: When $t = 0$

$$x = x_0, \text{ and } dx/dt = 0 \quad [d]$$

and when $t = \pi/\omega$

$$x = -x_0, \text{ and } dx/dt = 0 \quad [e]$$

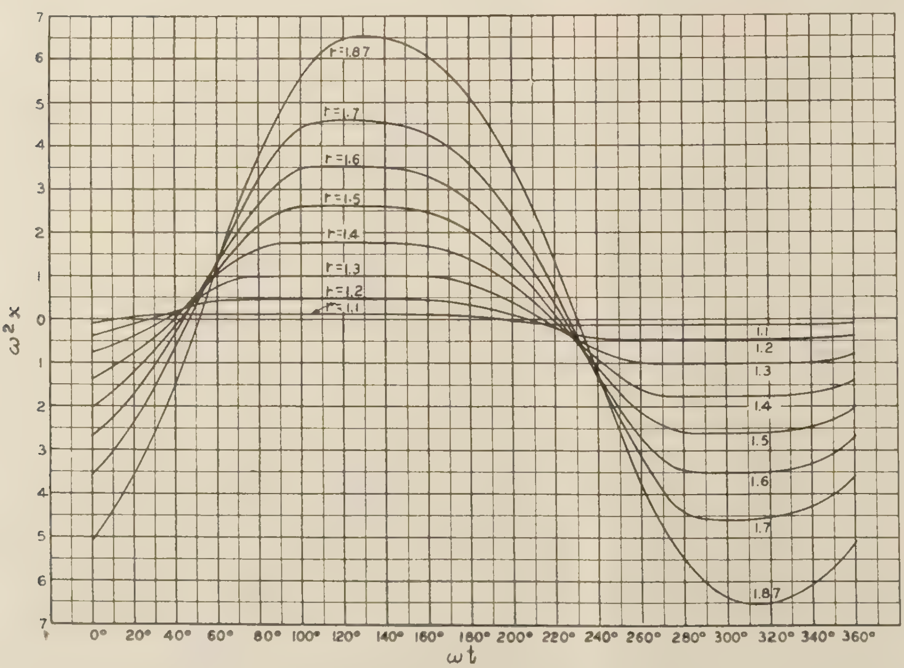

Fig. 16 Lateral Sliding of a Mass (Vehicle) on a Dry Surface (Rail) Under the Action of a Harmonic Force (Roll of Sprung Mass)

(r = maximum exciting force/friction force.)

By using Equation [b] and condition [d], C' may be solved for, whence

$$C' = -(D/m\omega) \sin \phi \ldots\ldots\ldots\ldots [f]$$

By using Equation [c] and condition [d], C'' may be solved for, whence

$$C'' = x_0 + (D/m\omega^2) \cos \phi \ldots\ldots\ldots\ldots [g]$$

By using Equations [b] and [f] and condition [e] we can obtain

$$\sin \phi = P\pi/2D \ldots\ldots\ldots\ldots [h]$$

By using Equations [c], [f], [g], and [h] and condition [e] we may solve for x and obtain the final solution

$$x = \frac{Pt}{2m}\left(t - \frac{\pi}{\omega}\right) + \frac{P}{2m\omega^2}\left(\sqrt{\frac{4D^2}{P^2} - \pi^2} \cos \omega t + \pi \sin \omega t\right) \ldots\ldots [i]$$

Now consider the case where the motion is intermittent, Fig. 15. During motion of the mass in one direction Equations [a], [b], and [c] still hold. However, the end conditions now, when $t = 0$, are

$$x = x_0,\ dx/dt = 0, \text{ and } d^2x/dt^2 = 0 \ldots\ldots\ldots\ldots [j]$$

and when $t = t_0$

$$x = -x_0, \text{ and } dx/dt = 0 \ldots\ldots\ldots\ldots [k]$$

We find that our values of C' and C'' are the same as before and are given by Equations [f] and [g]. By using Equation [a] and condition [j] we find that

$$\cos \phi = -P/D \ldots\ldots\ldots\ldots [l]$$

Now substitute condition [k] into Equation [b], using the known values of the constants for which we have already solved. We obtain

$$\omega t_0 - \sin \omega t_0 + \sqrt{(D^2/P^2) - 1}\,(\cos \omega t_0 - 1) = 0 \ldots [m]$$

Using the same procedure with Equation [c] instead of [b] we obtain

$$x_0 = \frac{t_0}{2m\omega}\sqrt{D^2 - P^2} - \frac{Pt_0^2}{4m} - \frac{P}{2m\omega^2}\left(\cos \omega t_0 + \sqrt{\frac{D^2}{P^2} - 1} \sin \omega t_0 - 1\right) \ldots\ldots [n]$$

The procedure should now be to solve [m] for t_0, substitute the value obtained into Equation [n], and assume that all of our unknown constants had been evaluated. Unfortunately, however, Equation [m] cannot be solved explicitly for t_0, so the best analytical solution we can give is that

$$x = \frac{Pt}{2m}\left(t - \frac{2}{\omega}\sqrt{\frac{D^2}{P^2} - 1}\right) + \frac{P}{m\omega^2}\left(\cos \omega t - 1 + \sqrt{\frac{D^2}{P^2} - 1} \sin \omega t\right) + x_0 \ldots\ldots [o]$$

where x_0 is given by Equations [n] and [m].

For any practical problem, however, the solution can be completed graphically. This has been done for several values of the ratio D/P, and the results are shown in Fig. 16. This chart may be used to find the lateral sliding of the wheel base for any D/P ratio, and no use need be made of the equation given in this Appendix.

It is of interest to know the value of D/P at which the motion changes from intermittent to continuous. This may be found by putting $\omega t_0 = \pi$ in Equation [m]. Solving for D/P we find

$$D/P = (1/2)\sqrt{(\pi^2 + 4)} = 1.87 \ldots\ldots [h]$$

Appendix 2

Values of $\sqrt{[x^2_i + (s/2)^2]}$ for use in applying Equation [9]

x_i	$\sqrt{x^2_i + (s/2)^2}$	$\sqrt{x^2_i + (s/2)^2} - x_i$	x_i	$\sqrt{x^2_i + (s/2)^2}$	$\sqrt{x^2_i + (s/2)^2} - x_i$	x_i	$\sqrt{x^2_i + (s/2)^2}$	$\sqrt{x^2_i + (s/2)^2} - x_i$	x_i	$\sqrt{x^2_i + (s/2)^2}$	$\sqrt{x^2_i + (s/2)^2} - x_i$
0	29.5	29.5	40	49.7	9.7	80	85.3	5.3	120	123.5	3.5
1	29.5	28.5	41	50.5	9.5	81	86.2	5.2	121	124.5	3.5
2	29.6	27.6	42	51.3	9.3	82	87.1	5.1	122	125.5	3.5
3	29.7	26.7	43	52.2	9.2	83	88.1	5.1	123	126.5	3.5
4	29.8	25.8	44	53.0	9.0	84	89.0	5.0	124	127.5	3.5
5	29.9	24.9	45	53.8	8.8	85	90.0	5.0	125	128.4	3.4
6	30.1	24.1	46	54.7	8.7	86	90.9	4.9	126	129.4	3.4
7	30.3	23.3	47	55.5	8.5	87	91.9	4.9	127	130.4	3.4
8	30.6	22.6	48	56.3	8.3	88	92.8	4.8	128	131.4	3.4
9	30.8	21.8	49	57.2	8.2	89	93.7	4.7	129	132.3	3.3
10	31.2	21.2	50	58.1	8.1	90	94.7	4.7	130	133.3	3.3
11	31.5	20.5	51	58.9	7.9	91	95.7	4.7	131	134.3	3.3
12	31.9	19.9	52	59.8	7.8	92	96.6	4.6	132	135.3	3.3
13	32.2	19.2	53	60.7	7.7	93	97.6	4.6	133	136.2	3.2
14	32.7	18.7	54	61.5	7.5	94	98.5	4.5	134	137.2	3.2
15	33.1	18.1	55	62.4	7.4	95	99.5	4.5	135	138.2	3.2
16	33.6	17.6	56	63.3	7.3	96	100.4	4.4	136	139.1	3.1
17	34.1	17.1	57	64.2	7.2	97	101.4	4.4	137	140.1	3.1
18	34.6	16.6	58	65.1	7.1	98	102.3	4.3	138	141.1	3.1
19	35.1	16.1	59	66.0	7.0	99	103.3	4.3	139	142.1	3.1
20	35.6	15.6	60	66.9	6.9	100	104.3	4.3	140	143.1	3.1
21	36.2	15.2	61	67.8	6.8	101	105.2	4.2	141	144.1	3.1
22	36.8	14.8	62	68.7	6.7	102	106.1	4.1	142	145.0	3.0
23	37.4	14.4	63	69.6	6.6	103	107.1	4.1	143	146.0	3.0
24	38.0	14.0	64	70.5	6.5	104	108.1	4.1	144	147.0	3.0
25	38.7	13.7	65	71.4	6.4	105	109.0	4.0	145	148.0	3.0
26	39.3	13.3	66	72.3	6.3	106	110.0	4.0	146	149.0	3.0
27	40.0	13.0	67	73.2	6.2	107	111.0	4.0	147	149.9	2.9
28	40.7	12.7	68	74.1	6.1	108	112.0	4.0	148	150.9	2.9
29	41.4	12.4	69	75.0	6.0	109	112.9	3.9	149	151.9	2.9
30	42.0	12.0	70	76.0	6.0	110	113.9	3.9	150	152.9	2.9
31	42.8	11.8	71	76.9	5.9	111	114.9	3.9	151	153.9	2.9
32	43.5	11.5	72	77.8	5.8	112	115.8	3.8	152	154.8	2.8
33	44.3	11.3	73	78.7	5.7	113	116.8	3.8	153	155.8	2.8
34	45.0	11.0	74	79.7	5.7	114	117.7	3.7	154	156.8	2.8
35	45.8	10.8	75	80.6	5.6	115	118.6	3.6	155	157.8	2.8
36	46.5	10.5	76	81.5	5.5	116	119.6	3.6	156	158.8	2.8
37	47.3	10.3	77	82.5	5.5	117	120.5	3.5	157	159.7	2.7
38	48.1	10.1	78	83.4	5.4	118	121.5	3.5	158	160.7	2.7
39	48.9	9.9	79	84.3	5.3	119	122.5	3.5	159	161.7	2.7

x_i	$\sqrt{x_i^2 + (s/2)^2}$	$\sqrt{x_i^2 + (s/2)^2} - x_i$	x_i	$\sqrt{x_i^2 + (s/2)^2}$	$\sqrt{x_i^2 + (s/2)^2} - x_i$	x_i	$\sqrt{x_i^2 + (s/2)^2}$	$\sqrt{x_i^2 + (s/2)^2} - x_i$	x_i	$\sqrt{x_i^2 + (s/2)^2}$	$\sqrt{x_i^2 + (s/2)^2} - x_i$
160	162.7	2.7	212	214.0	2.0	264	265.6	1.7	316	317.4	1.4
161	163.7	2.7	213	215.0	2.0	265	266.6	1.7	317	318.4	1.4
162	164.7	2.7	214	216.0	2.0	266	267.6	1.7	318	319.4	1.4
163	165.6	2.6	215	217.0	2.0	267	268.6	1.7	319	320.4	1.4
164	166.6	2.6	216	218.0	2.0	268	269.6	1.6	320	321.4	1.4
165	167.6	2.6	217	219.0	2.0	269	270.6	1.6	321	322.4	1.4
166	168.6	2.6	218	220.0	2.0	270	271.6	1.6	322	323.3	1.3
167	169.6	2.6	219	221.0	2.0	271	272.6	1.6	323	324.3	1.3
168	170.6	2.6	220	222.0	2.0	272	273.6	1.6	324	325.3	1.3
169	171.6	2.6	221	223.0	2.0	273	274.6	1.6	325	326.3	1.3
170	172.5	2.5	222	224.0	2.0	274	275.6	1.6	326	327.3	1.3
171	173.5	2.5	223	224.9	1.9	275	276.6	1.6	327	328.3	1.3
172	174.5	2.5	224	225.9	1.9	276	277.6	1.6	328	329.3	1.3
173	175.5	2.5	225	226.9	1.9	277	278.6	1.6	329	330.3	1.3
174	176.5	2.5	226	227.9	1.9	278	279.6	1.6	330	331.3	1.3
175	177.5	2.5	227	228.9	1.9	279	280.6	1.6	331	332.3	1.3
176	178.5	2.5	228	229.9	1.9	280	281.6	1.6	332	333.3	1.3
177	179.4	2.4	229	230.9	1.9	281	282.5	1.5	333	334.3	1.3
178	180.4	2.4	230	231.9	1.9	282	283.5	1.5	334	335.3	1.3
179	181.4	2.4	231	232.9	1.9	283	284.5	1.5	335	336.3	1.3
180	182.4	2.4	232	233.9	1.9	284	285.5	1.5	336	337.3	1.3
181	183.4	2.4	233	234.9	1.9	285	286.5	1.5	337	338.3	1.3
182	184.4	2.4	234	235.9	1.9	286	287.5	1.5	338	339.3	1.3
183	185.4	2.4	235	236.8	1.8	287	288.5	1.5	339	340.3	1.3
184	186.4	2.4	236	237.8	1.8	288	289.5	1.5	340	341.3	1.3
185	187.3	2.3	237	238.8	1.8	289	290.5	1.5	341	342.3	1.3
186	188.3	2.3	238	239.8	1.8	290	291.5	1.5	342	343.3	1.3
187	189.3	2.3	239	240.8	1.8	291	292.5	1.5	343	344.3	1.3
188	190.3	2.3	240	241.8	1.8	292	293.5	1.5	344	345.3	1.3
189	191.3	2.3	241	242.8	1.8	293	294.5	1.5	345	346.3	1.3
190	192.3	2.3	242	243.8	1.8	294	295.5	1.5	346	347.3	1.3
191	193.3	2.3	243	244.8	1.8	295	296.5	1.5	347	348.3	1.3
192	194.3	2.3	244	245.8	1.8	296	297.5	1.5	348	349.3	1.2
193	195.2	2.2	245	246.8	1.8	297	298.5	1.5	349	350.2	1.2
194	196.2	2.2	246	247.8	1.8	298	299.5	1.5	350	351.2	1.2
195	197.2	2.2	247	248.8	1.8	299	300.5	1.5	360	361.2	1.2
196	198.2	2.2	248	249.8	1.8	300	301.5	1.5	370	371.1	1.1
197	199.2	2.2	249	250.7	1.7	301	302.4	1.4	380	380.1	1.1
198	200.2	2.2	250	251.7	1.7	302	303.4	1.4	390	391.1	1.1
199	201.2	2.2	251	252.7	1.7	303	304.4	1.4	400	401.0	1.0
200	202.2	2.2	252	253.7	1.7	304	305.4	1.4	425	426.0	1.0
201	203.2	2.2	253	254.7	1.7	305	306.4	1.4	450	450.9	0.9
202	204.1	2.1	254	255.7	1.7	306	307.4	1.4	475	475.9	0.9
203	205.1	2.1	255	256.7	1.7	307	308.4	1.4	500	500.8	0.8
204	206.1	2.1	256	257.7	1.7	308	309.4	1.4	550	550.8	0.8
205	207.1	2.1	257	258.7	1.7	309	310.4	1.4	600	600.7	0.7
206	208.1	2.1	258	259.7	1.7	310	311.4	1.4	700	700.6	0.6
207	209.1	2.1	259	260.7	1.7	311	312.4	1.4	800	800.5	0.5
208	210.1	2.1	260	261.7	1.7	312	313.4	1.4	900	900.4	0.4
209	211.1	2.1	261	262.7	1.7	313	314.4	1.4	1000	1000.4	0.4
210	212.1	2.1	262	263.7	1.7	314	315.4	1.4	....	......	...
211	213.1	2.1	263	264.7	1.7	315	316.4	1.4	....	......	...

Appendix 3

Effect of Nonsinusoidal Lateral Motion of Wheel Base

The theory described in this paper assumes that all displacements, velocities, and accelerations vary harmonically. This assumption is reasonable for the motions of the sprung weight on its supporting springs, since the scales of the springs are linear and the predominating forces are spring forces and inertia forces. Also, accelerometer records taken in the cabs of oscillating locomotives appear sinusoidal, which shows that the velocities and displacements must have been very close to following pure sine waves. The unsprung mass, however, has its lateral motion resisted partly by friction and partly by flange pressure, the latter coming into play only near the maximum displacement positions. Therefore, it cannot be assumed that the lateral motions of the unsprung mass will be harmonic. Let us determine the effect that nonharmonic motion will have on the theory.

At the critical speed, the energy input per cycle is

$$4 \int_0^{\pi/2\omega} 4\mu QV\alpha dt = 16\mu QV \int_0^{\pi/2\omega} \alpha dt \ldots\ldots [a]$$

The energy loss per cycle is

$$8BA\mu Q \ldots\ldots [b]$$

Equating input to loss and solving for V, we obtain

$$V_{cr} = BA/2 \int_0^{\pi/2\omega} \alpha dt \ldots\ldots [c]$$

The determination of the critical speed is thus dependent on the determination of an expression for α as a function of t. If, as a first approximation we take $\alpha = A \cos \omega t$, as done before, we obtain

$$V_{cr} = 2B\omega/\pi \ldots\ldots [d]$$

Consider that α is some function other than $A \cos \omega t$. The turning action on the vehicle comes from the flanges, which act during only a portion of each cycle. Therefore, the angularity may change rather quickly, remain constant, and then change back again. The angularity α plotted against the time t Fig. 17 will be a sine wave flattened at the peaks. The limit of such a tendency is the square wave form, Fig. 17, which could not exist, but is a limit which could be approached. It represents the most unfavorable condition because it produces the largest possible value of input.

If the wave form of α were square, then α would be constant and equal to $\pm A$ during each half cycle. Substituting A for α in Equation [c] of this Appendix, we obtain

$$V_{cr} = B\omega/2 \ldots\ldots [e]$$

If the wave form of the variation of α were sharper at the peaks than a sine wave, then the input would be smaller and the critical speed would be higher. However, this is too favorable an assumption. If the tendency exists at all it comes as a result of the speed and flange pressure being too small to turn the locomotive fast enough. High speeds and severe oscillations both tend to produce the square rather than the pointed wave form.

As a result of the foregoing analysis we find that the critical-speed theory advanced in this paper no longer depends on the

assumption of sinusoidal motion. If the motion of the wheel base is not sinusoidal, the critical speed may be reduced by the factor $\pi/4$ or 0.785 as the limit.

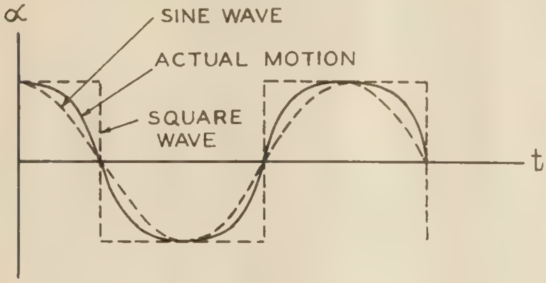

FIG. 17

Appendix 4

Nomenclature and Important Equations

α = the variable angle between the center line of the vehicle and the center line of the track

A = the peak value of α during a cycle of the oscillation

B = the wheel base of a vehicle

b = the lateral distance between the springs supporting the sprung weight of a rail vehicle

C' and C'' = constants of integration

c = track damping constant

D = disturbing force (see Appendix 1)

d = lateral displacement of the center of gravity of the sprung weight relative to the unsprung weight due to roll

f = natural frequency of roll of a vehicle

G = the lateral component of the friction force at the tread which opposes rotation of the wheel base

h = height of the center of gravity of the sprung mass above the center line of the axle

I_r = the polar moment of inertia of the sprung weight about a longitudinal axis through the axle centers

J = the polar moment of inertia of a vehicle about a vertical axis through the center of gravity

i = a general subscript designating the i-th axle of a wheel base

K_s = the spring scale for one side of a vehicle, neglecting cross-equalized springs

K = the lateral spring scale of a vehicle

k = axle load/average axle load

M = the mass of the sprung weight of a vehicle

μ = the coefficient of friction

n = the number of axles in a wheel base

ϕ = the phase angle between the lateral slippage of the wheel base and the rolling of the sprung weight (see Appendix 1)

P = the frictional resistance to the lateral slippage of the wheel base (see Appendix 1)

Q = static wheel load

s = track gage

δ = half the maximum lateral amplitude of the wheel base allowed by the track and bearing clearances

θ_i = phase angle of motion of i-th axle referred to phase of center of rotation

t = time

V = the forward velocity of the vehicle

ω = 2π times the frequency of the oscillation

W = the sprung weight of a vehicle

x = lateral slippage of wheel base due to roll of body (see Appendix 1)

x_i = the longitudinal distance between the i-th axle and the center of rotation

x_1 = same for the first axle

Y = the flange pressure on an axle

y = lateral translation of a vehicle.

Natural frequency of roll

$$f = [1/2\pi]\sqrt{[(K_s b^2 - 2Wh)/2I_r]} \qquad [10]$$

Angularity of wheel base

$$A = \delta\omega\cos\theta_1/V = \delta\omega/\sqrt{(x_1^2\omega^2 + V^2)} \qquad [14] \text{ and } [16]$$

Amplitude of lateral translation of wheel base

$$2y = 2VA/\omega \qquad [11]$$

Total amplitude of the i-th axle

$$= 2VA/\omega \cos\theta_i \qquad [12]$$

Input per cycle

$$= 2\pi n\mu QV\delta/\sqrt{(x_1^2\omega^2 + V^2)} \qquad [18]$$

Loss per cycle

$$= [8\mu Q\omega\delta/\sqrt{(x_1^2\omega^2 + V^2)}] \sum_{i=n}^{i=1} \sqrt{[x_i^2 + (s/2)^2]} \qquad [19]$$

Critical speed

$$V_{cr} = [8f/n] \sum_{i=n}^{i=1} \sqrt{[x_i^2 + (s/2)^2]} \qquad [9]$$

Critical speed for nonuniform axle load

$$V_{cr} = 8f\left[\frac{k_1\sqrt{[x_1^2 + (s/2)^2]} + k_2\sqrt{[x_2^2 + (s/2)^2]} + \ldots}{k_1 + k_2 + \ldots}\right] \qquad [20]$$

Measurement of Steam Rate and Indicated Horsepower of Locomotives

By ARTHUR WILLIAMS,[1] EAST CHICAGO, IND.

It has been the customary procedure to determine the steam rate and indicated horsepower of locomotives and other reciprocating steam engines from indicator cards. This paper points out that particularly at high locomotive speeds, the obtainment of indicator cards is subject to gross errors, and that the expense of rigging up for such tests is hardly warranted by the results that can be expected. The paper discusses in detail another method of determining the steam rate and indicated horsepower of a locomotive, which other writers have referred to as the "heat-drop method." The paper proves both the simplicity of the method as well as its relative accuracy. The method is applicable to such locomotives only in which the exhaust steam is at least dry, and only in the absence of moisture, the Btu contents per pound of both admitted and exhausted steam can be accurately determined through pressure and temperature measurement. The paper contains detailed suggestions for successfully applying the method on the locomotive. Thus it contains data on the construction and location of the thermocouples, and the manner in which these thermocouples are applied and wired up to the instrument with which the pyroelectric force is determined. The results of the tests carried out with this method on two different locomotives, are presented in the paper and, in general, it is shown what advantages this simple method has for the test and analysis of all those design factors or devices which are liable to affect the steam rate of a locomotive.

Toward the conclusion of the paper, reference is made to a suggestion by L. K. Botteron, which appeared in the *Railway Mechanical Engineer* of July, 1930, to the effect that the exhaust nozzle of a locomotive could be considered a flowmeter nozzle for the determination of the quantity of steam that the locomotive exhausts. The paper proceeds to show how the specific volume and velocity of the steam, while passing through the mouth of the locomotive exhaust tip (which quantities are essential for the Botteron method), are derived from the exhaust-steam temperature and pressure measurements that are incidental to the carrying out of the heat-drop determination.

The overall efficiency of a locomotive may be divided into three parts: The engine or cylinder efficiency, the boiler-combustion efficiency, and the machine efficiency. A measure of the cylinder efficiency is the steam rate, or steam consumption per indicated horsepower-hour. In the past this has been determined by measuring the indicated horsepower, and the steam to the engines, and dividing the second by the first. It is somewhat difficult on a road test to determine these values accurately. The indicated horsepower can be calculated when the mean effective pressure in the cylinders and the speed are known. The mean effective pressure in the cylinders is found by means of an indicator. The accuracy of the indicator depends in the first place upon the accuracy of the reducing motion from the crosshead to the indicator drum, the pencil motion on the indicator, the spring, and the area of the piston. With careful workmanship and calibration these errors can be made fairly small. It is more difficult, particularly at high speeds, to guard against errors due to the inertia of the moving parts and vibration of the apparatus.

Having obtained the mean effective pressure it is necessary to know the speed in order to calculate the indicated horsepower. A speed indicator is again a source of error, and if speeds are obtained by taking the time between mileposts the speed at the time at which the indicator card is taken may be different from the average time between mileposts.

The next step in the calculation of the steam rate is to measure the steam to the engines. This is usually done by measuring the water fed to the boiler from the tender tank and subtracting the steam to the auxiliaries, such as the air compressor, boiler feed pump, stoker, steam used for heating the train, or for other train services, and the steam wasted from the safety valve, injector overflow, and blower. Errors arise, of course, in the measurement or calculation of each of these quantities.

Since the accurate measurement of the steam rate depends upon the accurate measurement of the mean effective pressure, speed, boiler evaporation, steam to auxiliaries, and steam wasted, it can readily be seen that the correct determination of the steam rate on a road test, using the test procedure outlined, is difficult. The suggestion has been made[2] that the steam rate be determined by means of the observation of the difference in heat content of the steam in the steam pipe and the exhaust. It is the purpose of this paper to show that, with reasonable precautions this can be done on most locomotives with good accuracy and far less trouble than by using the indicator.

Theory of Measurement

The first law of thermodynamics states that "heat and mechanical energy are interconvertible and can neither be created nor destroyed. For steady-flow conditions, such as are obtained with an engine, it follows that "For any prime mover operating under these conditions the energy delivered by this apparatus in any unit of time is equal to the difference of the heat contents at entrance and exit from the apparatus, for the entire amount of

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

[2] "Some Experimental Results From a Three-Cylinder Compound Locomotive," by Lawford H. Fry, Proceedings, Institution of Mechanical Engineers, 1927, vol. 2, pp. 923–1024.

"A Thermodynamic Analysis of the Steady Flow of Fluids," by C. H. Berry, *Mechanical Engineering*, vol. 51, 1929, p. 816.

working substance flowing in this unit of time, minus the radiation and conduction losses from the apparatus."[3]

With superheated steam, if the temperature and pressure are known, the heat content can be obtained from steam tables. With saturated steam it is necessary to know the percentage of moisture. To measure the temperature of the exhaust steam from locomotive cylinders, when it is superheated, is simple. To measure the amount of moisture in the exhaust steam accurately, when it is saturated, is difficult, if not impossible. Accordingly, the method described in this paper is limited to those cases where there is superheat in the exhaust steam. Since most locomotives in main-line service have some superheat in their exhaust, this limitation is not very important. It is one, however, that must always be borne in mind.

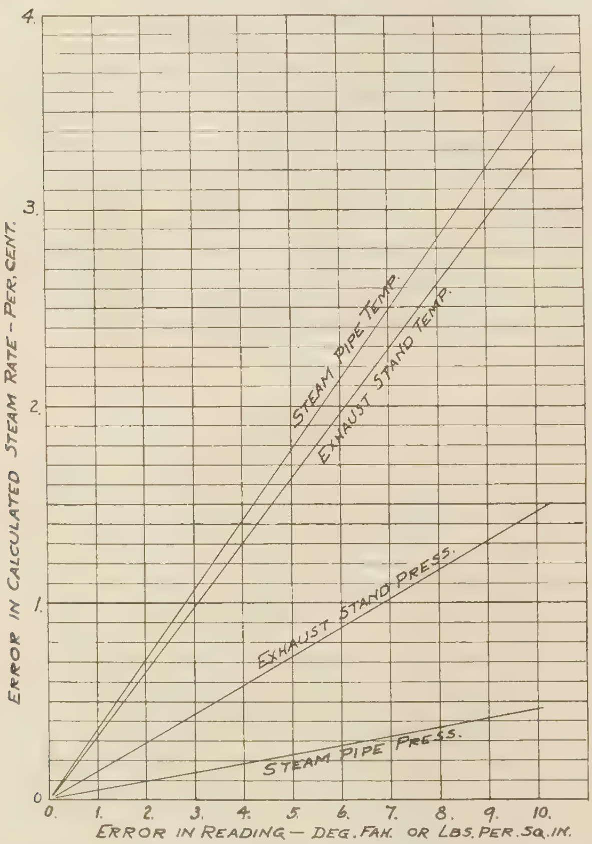

Fig. 1 Effect of Errors in Readings on Calculated Steam Rate

Knowing the temperature and pressure of the admitted steam and of the exhaust steam, the two heat contents can be obtained from steam tables. The difference between the two gives the heat drop in Btu per pound of steam. This heat drop is equal to the work done in the cylinders per pound of steam, plus the heat lost through radiation. The radiation loss is relatively small and, if desired, can be allowed for. One horsepower-hour is equal to 2545 Btu. Dividing 2545 by the heat drop per pound of steam will give directly the pounds of steam per indicated horsepower-hour. It makes no difference what processes are taking place in the cylinder, whether they are adiabatic, isothermal, or, as is actually the case, a turbulent process which does not follow any definite law. In the extreme case, where steam is leaking by the piston, with no work being done, the heat content would be the same in the steam pipe and in the exhaust, with due allowance for radiation. This case is the same as that of the well-known throttling calorimeter.

[3] "Heat-Power Engineering," Part 1, Thermodynamics and Prime Movers, by W. N. Barnard, F. O. Ellenwood, and C. F. Hirshfeld, John Wiley & Sons, Inc., New York, 1926, p. 46.

Value of Method for Locomotive Tests

Since it is only necessary to measure the temperature and pressure in the steam pipe and in the exhaust of the locomotive cylinders, the test apparatus is relatively simple. It is possible for one man to ride in the cab of a locomotive and take all of the necessary readings. This may be contrasted with the test apparatus and test crew necessary for determining the steam rate using indicators.

It is obvious that where a complete test by taking indicator cards would not be justified, a test using the heat-drop method can be run with a low expenditure, and the various design features which influence cylinder performance can be studied as often as desired.

The following are the design features commonly in question:

(1) Boiler pressure, since it controls the steam pressure in the steam pipe.

(2) The throttle and dry pipe. The pressure drop through these affects the pressure in the steam pipe.

(3) The steam temperature in the steam pipe and superheater design.

(4) The size and design of the valves and cylinders.

(5) The valve setting.

(6) The exhaust nozzle, both size and shape.

The necessary readings for obtaining the steam rate can be taken at frequent intervals, so that a complete picture is easily obtained of the locomotive performance, at various speeds and rates of working. Even when tests are being made on the combustion efficiency of, and heat transfer in, the boiler, it is desirable to measure the steam rate and cylinder efficiency, so that the results may be more closely analyzed.

Methods of Measurement and Sources of Error

The instruments used in measuring the temperature and pressure should be as accurate as possible and suitable for use in road tests on locomotives. In order to obtain a better understanding of the relation between errors and accuracy, Fig. 1 has been prepared. In this figure the four curves show the effect of errors in measurement of the steam-pipe temperature, steam-pipe pressure, exhaust-steam temperature, and exhaust-steam pressure, on the steam rate. An error of 2 lb per sq in. in measuring the steam-pipe pressure will cause an error of 0.1 per cent in the calculated steam rate. An error of 1 lb per sq in. in the measurement of the exhaust pressure will cause an error of 0.14 per cent in the steam rate. To measure the steam-pipe pressure to within 2 lb per sq in. and the exhaust pressure to within 1 lb per sq in. only calls for reasonably accurate test gages. It may be necessary to make a correction for the hydrostatic head, due to the water in the pipe connecting the pressure gage and the point where it is tapped into the steam pipe or exhaust pipe.

The measurement of the steam pipe and exhaust-steam temperatures calls for more careful consideration. If they can be measured with a maximum error in each of 1 F, the resulting error in the steam rate will be 0.68 per cent. If they are measured with a maximum error in each of 2 F, the error in the steam rate will be 1.35 per cent. Adding to these figures the errors due to the steam-pressure measurements gives a total error of 0.92 per cent if the temperatures are measured to within 1 F and 1.59 per cent if the steam temperatures are measured to within 2 F.

It is possible to use instruments suitable for road tests on locomotives that will give results which will average somewhere between these two figures.

The apparatus described in this paper has been used by the author with good results. This statement is not meant to infer that this is the only apparatus suitable for measuring temperatures to within 1 or 2 F. Other means of temperature measurement will be mentioned later.

The temperatures are measured with thermocouples constructed as shown in Fig. 2. Wires used are iron and constantan, purchased in lengths. Samples of each length are calibrated, with reference to a thermometer which has been certificated by the Bureau of Standards. The wires are led through porcelain insulators to a steel plug welded in the end of a piece of 1/4-in. pipe. The wires are brought through small holes drilled in the steel plug, bent over, and the ends then covered with silver solder. This gives a thermocouple which is tight against the steam pressure and will follow any change in steam temperature rapidly. The thermocouple is screwed into a socket, which, in turn, is screwed into the steam pipe. This gives a steam space surrounding the thermocouple all the way up to the thermocouple head, which serves to minimize any errors due to conduction along the thermocouple pipe. The steam-pipe thermocouple, illustrated in Fig. 2, is screwed into the steam pipe in any convenient place.

The exhaust-steam temperature is measured, not in the exhaust passage, but in the exhaust stand. It has been the common practice in the past to measure the exhaust-steam temperature in the exhaust passage close to the steam chest. This has been standard practice on the Pennsylvania Railroad in their tests, at Altoona, in their locomotive test plant. It is stated in sev-

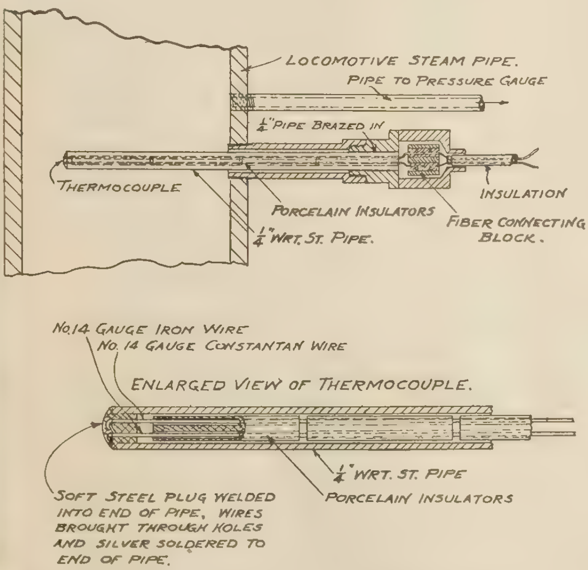

FIG. 2 STEAM-PIPE THERMOCOUPLE AND PRESSURE-GAGE CONNECTION

eral of the railroad's publications, that the exhaust-steam temperature measurements are believed to be higher than the true temperatures. The reason given is as follows:

> During admission of the steam into the cylinder, heat is transferred from the steam to the cylinder walls and head. As the steam expands and becomes cooler this heat transfer stops and then reverses, so that during the exhaust stroke heat is being transferred from the cylinder to the exhaust steam. After release, when the pressure in the cylinder is higher than the average exhaust pressure, there occurs a sudden rush of steam, which by reason of its high velocity does not have time to absorb very much heat from the cylinder walls. As the piston moves on the exhaust stroke the steam, which is now moving more slowly, is heated up by the cylinder walls, with a rise in temperature.

The Pennsylvania Railroad states that it believes the sudden rush of relatively cool steam at release does not register properly on the thermometer in the exhaust passage and that this thermometer is influenced more by the slow-velocity high-temperature steam during the exhaust stroke.

In measuring any fluctuating steam temperature it is desirable to provide for as much mixing of the steam as possible. Accord-

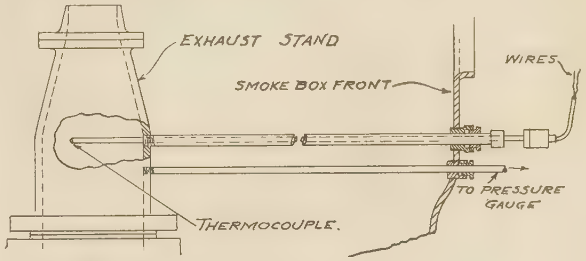

FIG. 3 EXHAUST-STAND THERMOCOUPLE AND PRESSURE-GAGE CONNECTION

ingly, the exhaust-steam thermocouple is located in the exhaust stand so that the steam passes through a chamber of some volume and with several bends before the temperature is measured. Also a thermocouple applied in one side of the exhaust stand will measure the temperature of the exhaust from both ends of one side of the engine, and thermocouples applied in both sides of the exhaust stand will measure the exhaust-steam temperatures from all four ends of the locomotive cylinders.

The accuracy of the assumption that better temperature measurements could be made in the exhaust stand than in the exhaust passage was tested on a locomotive in main-line service. On this particular test an observer was riding on the front of the locomotive and simultaneous readings were taken of the exhaust-steam temperature with a thermometer in the exhaust passage close to the steam chest, and a thermocouple in the exhaust stand. The average of a number of readings showed the observed exhaust-passage temperature, as indicated by the thermometer, to be 345 F and the observed exhaust-stand temperature, as indicated by the thermocouple, to be 325 F. In other words, if the exhaust-stand temperature is taken as being accurate the temperature measured in the exhaust passage close to the steam chest was reading 20 deg high. This figure would vary no doubt with the class of engine and conditions of working, but it is evident that measurements taken of the exhaust-steam temperature in the exhaust passage close to the steam chest are subject to considerable error.

The application of the thermocouple to the exhaust stand is shown in Fig. 3. The thermocouple is constructed in the same way as that shown in Fig. 2. A 1/2-in. standard pipe is screwed in the exhaust stand and brought out through the smokebox front. The thermocouple is slid into this 1/2-in. pipe, which provides an insulating steam jacket and also permits easy removal.

The thermocouple electromotive force is measured with a Leeds and Northrup potentiometer. The potentiometer is a well-known means of measuring temperatures in connection with thermocouples, and is suitable for locomotive testing. Because of severe vibration it is always possible to obtain a broken strand in one of the thermocouple wires, or a bad contact. With a direct-reading pyrometer this would throw the instrument off but with a potentiometer it would make no difference, as long as the wiring was good enough to provide a circuit.

With the galvanometer mounted in the potentiometer case as sent out by the makers, the vibration of the locomotive is sufficient to upset the galvanometer needle. In order to obviate this difficulty, the galvanometer is mounted on a board with a spring suspension as shown in Fig. 4. This board also serves for writing down the data and is held by the observer. The observer's body will absorb most of the shocks and vibration of the locomotive, and with the spring mounting of the galvanometer no trouble will be found in taking readings. The potentiometer is mounted on a bracket and the galvanometer wired to

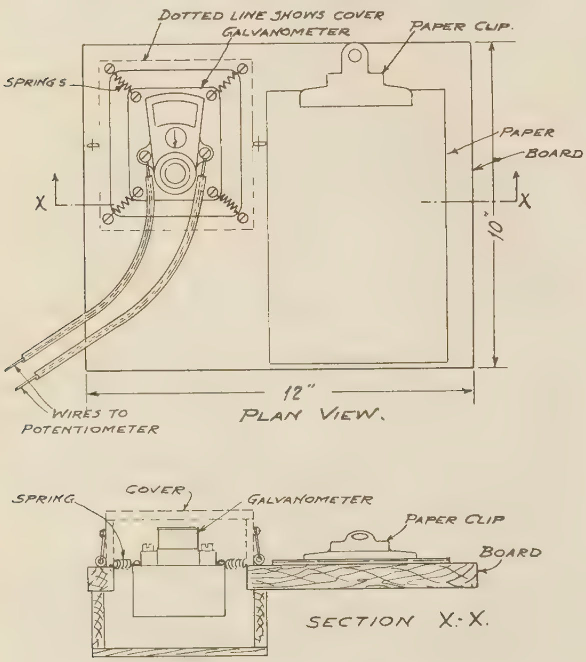

Fig. 4 Galvanometer Mounting of Data Board

the potentiometer by means of two copper wires of sufficient length to allow freedom of movement to the observer.

The thermocouples are connected to a rotary switch in the cab by means of which each one in turn may be put into the potentiometer circuit. The wiring diagram is shown in Fig. 5. A thermocouple measures the difference between the temperatures of the hot and cold junctions and consequently it is necessary to know the temperature of the cold junction and to connect the hot and cold junctions in such a way that no other electromotive force is set up. As in Fig. 5, the thermocouples are connected with special extension leads to a connecting strip which is located at some convenient spot near the front of the locomotive. These extension leads are of the same material as the thermocouple wire, so that no electromotive force is set up at their junction. This connecting strip is mounted in a closed metal box, so that all the junctions will be at the same temperature. From the connecting strip copper leads are run to the rotary switch in the cab. The common wires from the rotary switch, which are connected to each thermocouple in turn, are connected with copper leads forward to the connecting strip and from there connected with the special extension leads back to the cab. The ends of these leads are joined to copper wire, which is connected to the potentiometer. This junction of the extension lead and the copper wire forms the cold junction and is placed in a piece of $^3/_4$-in. pipe. The thermometer and cold junction will be at the same temperature and will not be affected by stray air currents. As many thermocouples as desired can be wired with extension leads to the connecting strip at the front of the engine and the single extension lead going back to the cold junction will automatically refer the cold junction from the connecting strip to the cab.

For best accuracy in measuring the steam rate it is desirable to have thermocouples in both steam pipes and both sides of the exhaust stand. The temperatures should not vary much from side to side, but with the two sets of thermocouples variations can be seen and a true average obtained.

Instead of using the thermocouples the temperatures can

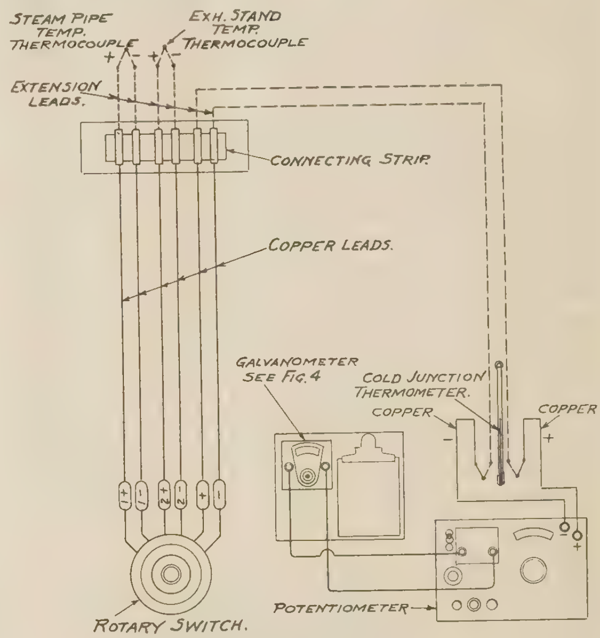

Fig. 5 Wiring Diagram

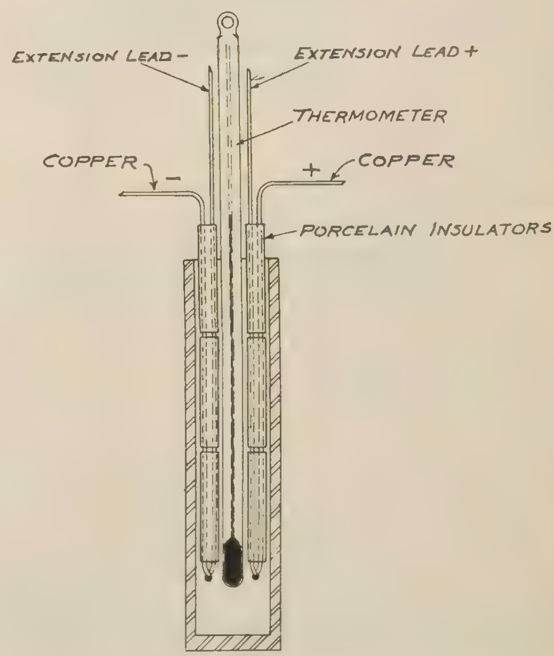

Fig. 6 Cold Junction

be measured with even greater accuracy by using platinum-resistance thermometers.

It has already been pointed out that the heat drop from the steam pipe to the exhaust includes radiation from the steam pipes, steam chests, cylinders, and exhaust passages. For a typical Pacific-type locomotive, this radiation loss has been calculated to be approximately 0.42 per cent of the heat drop with the engine standing still and 0.83 per cent of the heat drop with the engine running at 100 mph. This radiation loss can be allowed for, but it is obvious that inaccuracies in its determination will be negligible in its effect on the calculation of the steam rate.

Road Tests of Locomotives

Table 1 shows the results obtained on road tests of a 2-8-2-type locomotive in fast freight service. The object of the test was to determine the effect on the engine performance of a special design of superheater unit. Tests were first run with the standard type-*A* superheater in place. The special superheater was then installed in place of the type-*A* and further tests were run. No attempt was made to obtain special conditions with reference to the running. The engine was in regular service, pulling whatever trains were assigned to it with various engine crews. The locomotive would run for half a division, then take on water, and then finish the run to the end of the division. Each test, shown in Table 1, is the average of readings taken every two miles for about 40 to 50 miles, representing about half a division. After starting, several miles were allowed before readings were commenced, in order that conditions might become reasonably constant. A caboose was carried back of the locomotive as a test car. The thermocouples were all wired to the caboose, where the temperatures were read by an observer with the potentiometer and rotary switch. The wiring connection between the caboose and tender was arranged to be easily connected and disconnected. Temperatures were measured on the left side of the engine only.

TABLE 1 TEST OF 2-8-2-TYPE LOCOMOTIVE

Test no.	Tonnage	Speed mph	Throttle opening	Cutoff, per cent	Steam pressure: Steam pipe, lb per sq in.	Steam pressure: Exhaust, lb per sq in.	Temperature: Steam pipe, F	Temperature: Exhaust steam, F	Steam rate, lb per ihp-hr
1	3779	30.7	Full	44	149.9	11.2	641	318	17.59
2	3879	24.0	Full	48	149.0	12.6	639	327	18.13
3	2016	42.4	Full	40	147.7	12.5	597	291	18.62
4	3000	28.8	Full	49	145.7	12.2	644	325	17.80
5	3532	27.8	90%	48	144.2	11.4	628	318	18.34
6	2690	32.0	Full	43	142.5	12.7	635	325	18.21
7	2629	34.3	Full	42	142.9	12.5	633	320	18.09
8	2699	30.6	Full	46	149.2	13.0	625	314	18.41
9	2819	29.3	Full	44	144.4	13.3	621	306	18.22
10	3760	27.6	Full	45	154.8	12.6	641	318	17.58
11	4200	27.3	Full	46	155.7	12.6	632	306	17.40
12	3476	33.3	90%	42	147.3	12.5	642	327	17.93
Avg	3207	30.7	Full	45	148	12.4	632	316	18.03
13	3637	29.8	Full	49	159.2	13.0	707	357	15.80
14	3718	28.5	Full	49	157.7	12.8	724	365	15.39
15	3680	33.4	Full	47	165.5	9.7	710	354	15.68
16	3701	30.8	Full	47	162.7	12.0	683	348	16.69
17	2966	28.1	Full	49	164.0	13.0	716	367	15.87
18	2623	31.9	Full	47	162.3	14.2	705	379	16.97
19	2660	38.0	Full	47	156.7	14.3	692	352	16.40
20	3067	37.2	Full	45	164.7	11.6	689	334	15.84
21	3789	31.4	Full	47	161.1	12.4	680	347	16.92
22	....	32.0	Full	45	160.4	11.2	677	357	17.53
23	....	36.9	Full	43	163.7	10.0	666	347	17.71
24	....	37.6	Full	47	159.8	14.5	718	395	17.21
25	....	29.8	Full	45	159.4	12.4	712	375	16.48
26	2580	36.6	Full	46	157.0	14.1	680	362	17.62
27	3670	26.3	Full	55	156.8	14.0	690	367	17.21
28	3400	31.6	Full	49	155.3	13.3	676	355	17.61
29	3382	27.7	Full	48	160.3	11.3	693	354	16.48
30	4112	30.8	Full	48	162.7	11.9	681	351	17.02
31	3354	31.7	Full	48	162.7	13.5	690	357	16.79
32	2754	37.4	Full	43	161.9	11.8	675	350	17.38
Avg	3318	32.4	Full	47	160.7	12.6	693	359	16.73

TABLE 2 TEST OF 4-6-2 TYPE LOCOMOTIVE

Test no.	No. of cars	Tonnage	Cutoff per cent	Pressure: Steam pipe lb per sq in.	Pressure: Exhaust, lb per sq in.	Temperature: Steam pipe, F	Temperature: Exhaust F	Steam rate, lb per ihp-hr
1	5	571	25	202	11.6	700	356	16.47
2	5	573	25	205	11.6	715	354	15.60
3	5	573	27	207	12.5	709	343	15.41
4	5	571	25	210	12.9	705	355	16.19
5	6	644	25	207	12.4	691	339	16.18
6	6	636	26	205	13.2	711	359	16.00
7	5	569	25	205	10.9	699	337	15.61
8	5	558	25	211	12.5	719	360	15.69
Avg	..	587	25.4	207	12.2	706	350	15.89
9	6	643	25	213	11.4	748	360	14.37
10	6	643	26	211	11.5	749	359	14.28
11	6	644	25	214	10	748	356	14.21
12	7	715	26	213	13.6	749	371	14.79
13	6	643	25	210	13.0	748	362	14.40
14	6	643	26	209	13.3	745	358	14.38
Avg	..	655	25.5	212	12.1	748	361	14.41

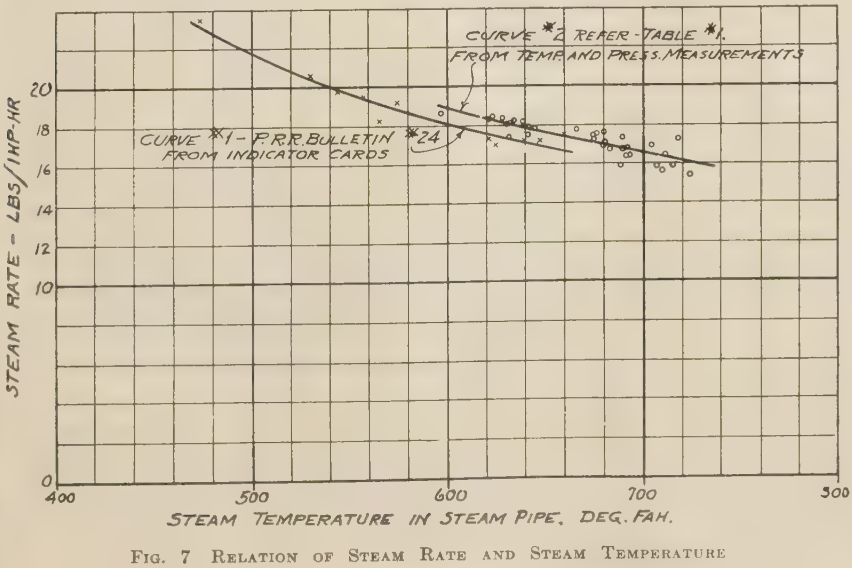

Fig. 7 Relation of Steam Rate and Steam Temperature

The method of testing was successful and the readings obtained were consistent and reasonable. The increase in steam temperature with the special type of superheater showed a decrease in steam rate, in line with other tests. This will be referred to later, when the accuracy of the heat-drop method is discussed.

Table 2 shows the results obtained in a similar test on a 4-6-2-type locomotive in fast passenger service. In this case temperatures were measured in both steam pipes and in both sides of the exhaust stand. The variation from side to side was not very great, but was enough to be significant. The potentiometer and rotary switch were mounted in the cab and even at the highest of speeds the galvanometer mounting shown in Fig. 4 was satisfactory. As in the test recorded in Table 1 the increase in steam-

pipe temperature showed a decrease in steam rate, as would be expected. The readings of the steam-pipe temperature, exhaust-stand temperature, and steam rate are more consistent than those shown in Table 1. This is because the weight of the train did not vary much. Only two crews were used for all of the tests, and the time-table schedule was kept more closely.

Accuracy of Heat-Drop Measurement of Steam Rate

It is difficult to estimate the absolute accuracy of the steam rate as determined by the proposed method. In a number of locomotive tests, particularly those made at the Altoona testing plant of the Pennsylvania Railroad, measurements were taken of the exhaust-steam temperature. This temperature was always measured in the exhaust-steam passage close to the steam chest, and, as already pointed out, the readings cannot be taken as being accurate. Consequently, it is not possible to compare the steam rate as obtained in these tests from the heat drop and, as obtained from the indicator cards, except in a general way.

Some idea of the possible accuracy of the heat-drop method can be obtained from Fig. 7. In this figure steam rate is plotted against temperature in the steam pipe. Curve 1 is taken from Pennsylvania Railroad bulletin No. 24, published in 1914, entitled "Superheater Tests." In this bulletin tests are described which were made on a Pacific-type locomotive. The only changes throughout the test were the length and arrangement of the superheater units. All points shown on curve 1 are for a constant cutoff of approximately 40 per cent and a consistent speed of approximately 240 rpm. The steam-pipe pressure varied from 170 to 195 lb. Curve 2 is plotted from the results given in Table 1. These tests were run at approximately 45 per cent cutoff, with a steam-pipe pressure of from 145 to 165 lb per sq in.

On account of the lower steam-pipe pressure it would be expected that curve 2 would lie above curve 1, showing a higher steam rate for the same steam temperature. The general trend of curve 2 is the same as that of curve 1. Another measure or indication of the accuracy of the results is the scattering of the points about the average line. It must be remembered that the tests for curve 1 were made on a stationary test plant, with the speed, cutoff, and horsepower being closely controlled. The tests for curve 2 were made in regular service, with a large variation in speed and tonnage, and with a number of different crews operating the locomotive. In view of this it is thought that the scattering of the points for curve 2 compares favorably with the scattering of the points for curve 1.

Measurement of Indicated Horsepower

It was pointed out by L. K. Botteron, in a paper in the *Railway Mechanical Engineer*, July, 1930, that the exhaust nozzle of a locomotive could be considered as a flowmeter. From the exhaust pressure and temperature the velocity of the steam in the exhaust-nozzle tip can be determined. Using the connections for pressure and temperature, as shown in Fig. 3, it is necessary to apply a correction for the velocity of the steam at the point where the pressure is measured. The usual formula for determining the velocity of steam in a nozzle is

$$V_2 = 223.7\sqrt{H_1 - H_2} \qquad [1]$$

where V_2 = outlet velocity, ft per sec
H_1 = inlet heat content, Btu per lb
H_2 = outlet heat content, Btu per lb.

In order to allow for the velocity of the steam at the point where the pressure is measured it is necessary to use the more exact formula

$$\frac{V_2^2 - V_1^2}{2g} = 778(H_1 - H_2) \qquad [2]$$

where V_1 = inlet velocity, ft per sec

From a knowledge of the pressure and temperature in the exhaust pipe the quantity H_1 is determined. From a Mollier diagram the quantity H_2 is obtained, since the steam expands adiabatically and the entropy is the same at points 1 and 2. The pressure at point 2 is the pressure in the smokebox, and from the entropy and pressure the specific volume can be determined. Since the same weight of steam is flowing by points 1 and 2, the relation between V_1 and V_2 is

$$V_1 = V_2 \times \frac{A_2}{A_1} \times \frac{v_1}{v_2} \qquad [3]$$

where A_1 = area of exhaust stand where pressure is measured, sq ft
A_2 = area of exhaust tip, sq ft
v_1 = specific volume at inlet, cu ft per lb
v_2 = specific volume at outlet, cu ft per lb.

Substituting this value of V_1 in Equation [2] gives

$$\frac{V_2^2\left[1 - \frac{A_2V_1}{(A_1v_2)^2}\right]}{2g} = 778\ (H_1 - H_2) \qquad [4]$$

from which V_2, or the velocity in the exhaust tip, can be calculated. The flow of steam through the exhaust tip is equal to the velocity multiplied by the area, multiplied by the density, multiplied by a coefficient. It was assumed in the article by Botteron that this coefficient would be unity. Using this assumption the author has obtained some results which agreed fairly well with the measurement of water from the tender tank. It is felt that for better accuracy it would be desirable to calibrate the exhaust tip by means of a standing or blowdown test. It would be possible for any railroad to establish the coefficient for their design of exhaust tip and then to use the exhaust-stand pressure and temperature measurements to give accurate readings of the flow of steam through the exhaust tip. By making suitable corrections for the steam flowing from the exhaust to the feedwater heater or exhaust-steam injector and for the steam flowing from the auxiliaries to the exhaust passage the weight of steam to the engine can be calculated with fair accuracy. These readings would give the flow of steam to the engines at any instant during a run. When the steam rate is known, as obtained from the heat-drop measurements, the indicated horsepower, at any instant, can be obtained by dividing the steam to the engines by the steam rate. This enables the indicated horsepower to be determined by measuring the temperature and pressure of the steam in the steam pipe and in the exhaust stand.

The Correlation of Spring-Wire Bending and Torsion Fatigue Tests

By E. E. WEIBEL,[1] BERKELEY, CALIF.

The investigation reported in this paper was an attempt to correlate bending and torsion fatigue results for tempered Swedish valve-spring wire so that the torsion fatigue strength might be inferred within reasonable limits of accuracy from the fatigue values found in the simpler rotating-beam test. The original program called for (*a*) the development of suitable methods of gripping the wire for the torsion fatigue tests; (*b*) the determination of bending and torsion fatigue values for one size of wire in the as-received condition, and also after removal of the surface layer in a centerless grinder; (*c*) the determination of fatigue values for other sizes of wire between $^1/_8$ in. and $^1/_4$ in., as permitted by time limitations; and (*d*) a survey of the literature for data bearing on the correlation of bending and torsion results. The results of these experiments on straight specimens of tempered Swedish wire in torsion and bending fatigue may be summarized as follows:

1 Methods of gripping were developed which are satisfactory for both bending and torsion fatigue tests with the exception of torsion tests on shot-blasted wire.

2 Zero to maximum torsion fatigue values for straight specimens in the as-received condition were about 70 per cent higher than values estimated from previous tests on carefully coiled springs.

3 The smaller of two sizes of wire tested in the as-received condition had the higher torsion fatigue limit and lower ultimate torsional strength.

4 Removal of the natural surface by grinding increased the bending fatigue strength about 20 per cent, but decreased the torsion fatigue strength about 4 per cent.

5 Shot-blasting increased the bending fatigue strength about 27 per cent and increased the torsion fatigue strength an undetermined amount.

6 The zero-to-maximum torsion fatigue limits of two sizes of wire tested in the as-received condition were 1.5 and 1.6 times their respective bending fatigue limits.

7 For two sizes of wire in the as-received condition the values of 0.84 and 0.90 were obtained for the ratio of reversed torsion fatigue strength to reversed bending fatigue strength.

8 Approximate measurements of surface residual stress were made on the wire in different conditions before and after fatigue testing and before and after coiling, which may have a bearing on the fatigue properties of springs.

1—REVIEW OF THE LITERATURE

Methods of Gripping Wire in Torsion Fatigue Tests

A DESCRIPTION of gripping methods satisfactory for torsion fatigue tests on tempered carbon-steel wire could not be found.

F. C. Lea of Sheffield has developed methods of gripping which are satisfactory for cold-drawn wires. In his earlier work (1)[2] split bushings were used. These were not entirely satisfactory for tempered wires. In his later work (2) gripping shanks of an alloy with a low melting point (type metal) were cast on the ends of the wire specimen. It was found that 80 per cent of the cold-drawn-wire specimens failed at a distance from the grips.

Goodacre (3) reviews gripping methods in use in England for bending, tension, and torsion fatigue tests on ferrous and nonferrous wires. In the case of torsion Lea's casting method is mentioned.

A study of type-metal alloys by F. D. Weaver (4) was noted; also a paper by L. J. G. van Ewijk (5) on the penetration of steel by soft solder and other molten metals at temperatures up to 400 C. The latter's observations may explain why Lea's casting method, unmodified, was not successful with tempered Swedish wire.

In 1929, O. Föppl (6) obtained increased torsion and bending fatigue strengths of polished specimens by rolling the surface so as to produce local plastic flow.

A publication by P. Behrens (7) describes the rolling apparatus used by Föppl, and gives further results. An increase of 17 per cent in torsion fatigue strength of a mild steel was obtained, and increases of 20 to 50 per cent in bending fatigue strength of a low-carbon steel.

Thum and Wunderlich (8) determined the bending fatigue strength of shafts with and without a press-fitted collar. Heavy stress concentrations, microscopic rubbing with the formation of oxides and corrosion, lowered the fatigue strength considerably. By surface-rolling the shaft and shaping the collar to reduce stress concentration, an increase of 95 per cent in fatigue strength was obtained.

A paper by O. J. Horger (9) presents a considerable amount of data on the surface-rolling process. Increases in bending

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[2] Numbers in parentheses correspond to similarly numbered references given in the bibliography at the end of the paper.

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

fatigue strength around 30 and 40 per cent were obtained. Petersen and Wahl (10) also experimented with surface-rolling. Their paper includes a summary of Thum and Wunderlich's results.

Fatigue Results on Tempered Swedish Valve-Spring Wire

Swan, Sutton, and Douglas (11) tested straight specimens of a tempered Swedish valve-spring wire in torsion fatigue with maximum stress four times the minimum stress. When tested at full diameter of 0.128 in. the safe range was from 20,200 to 80,800 lb per sq in. When reduced to 0.100 in. diameter and polished, the safe range was from 31,400 to 125,500 lb per sq in., an increase of 56 per cent. These stress values were not measured directly but were assumed proportional to the angular travel of the gripping chucks, an assumption which will be shown later to be unjustified because of variations in the effective length of the specimen.

F. P. Zimmerli (12) tested a number of steels in the form of completed springs. The steels, numbers 4 and 7, were straight carbon pretempered Swedish wires, 0.148 in. diameter, with 0.75 and 0.65 per cent carbon content, respectively. The safe ranges decrease with increasing mean values of torsion stress in close agreement with the Goodman diagram. Probable values of zero to maximum torsion stress range extrapolated from the diagrams are 68,000 and 58,000 lb per sq in., respectively.

Bending fatigue tests were made on two sizes of Swedish valve-spring wire by Shelton and Swanger (13). The first wire, 0.162 in. diameter, 0.65 per cent carbon, had an ultimate tensile strength of 221,000 lb per sq in. and with the natural surface a fatigue strength 34 per cent of the ultimate, or $\pm 76{,}000$ lb per sq in. When reduced to 0.130 in. diameter and polished, the fatigue strength was 56 per cent of the ultimate or $\pm 126{,}000$ lb per sq in. The second wire, 0.148 in. diameter, 0.65 per cent carbon, had an ultimate strength of 217,000 lb per sq in. and a fatigue strength only 30 per cent of the ultimate or $\pm 65{,}000$ lb per sq in. It showed surface defects.

Other Fatigue Results

The torsion fatigue results of G. A. Hankins (14) on polished specimens of spring steel show safe stress ranges which are practically the same for alternating torsion and for zero-to-maximum torsion stress. For higher mean stresses there is a reduction in the safe range but values are always greater than would be given by the Goodman diagram.

The Correlation of Torsion and Bending Fatigue Results for Wire

No reports of experiments were found in which torsion and bending fatigue results were obtained for the same wire and compared. Considerable experimental work and some theoretical work having a bearing on the torsion fatigue strength and on the ratio of torsion to bending fatigue strengths was reviewed.

A criterion of fatigue failure of metals under any type of stress does not appear to have been found. Fahrenhorst and Schmid (15) review briefly the hypotheses as to the cause of fatigue failure of Kuntze, Haigh, Ljungberg, Ono, Ludwik, Gough, Hanson and Wright, and Schmid. These are many and various and will not be given here.

H. J. Gough (16) reported that behavior of metals was being studied under reversed bending, reversed torsion, and all possible combinations of these two types of fatigue loading. Mild steel, a ductile material, obeyed closely the distortion-energy theory of von Mises and Hencky and discussed in a paper by Nadai (17), according to which theory the safe torsion fatigue stress would be 0.577 times the safe bending fatigue stress. But an alloy steel and a cast iron exhibited other relations. He concludes that no general criterion of failure obtains under the stated stress conditions.

Fatigue experiments by A. F. Maier (18) on tubes under two types of reversed stressing obtained by means of internal pressure appeared to support the theory of maximum shear stress according to which the safe shear stress is 0.500 times the safe tensile stress. The results did not support the distortion-energy theory.

Moore and Kommers (19) list the reversed-torsion endurance limits and the reversed-bending endurance limits for polished specimens of a number of straight-carbon steels which had received widely different heat-treatments. The average of 19 values of the ratio of torsion endurance limit to bending endurance limit was 0.55. Extreme values were 0.48 and 0.64. The average of 49 values of the same ratio, from results of tests on polished specimens, listed by Gough (20), was 0.56. Extreme values were 0.44 and 0.80. These results are for presumably homogeneous and isotropic material, and surface effects have been eliminated as far as possible.

Gill and Goodacre (21) studied some aspects of the fatigue properties of patented wire. They found that wire with decarburized surface will take overloads as well as, or better than, wire not decarburized and conclude that there is no gain in removing the decarburized surface if wire will be overstressed much. The writers assume that the ratio of torsion fatigue strength to bending fatigue strength should have the value $^2/_3$, and using this value together with one of their bending fatigue results for polished wire, they calculate a hypothetical torsion fatigue strength which agrees closely with a previous experimental torsion fatigue value for polished wire obtained by Lea and Dick (22). They say, "It seems very probable therefore that the fatigue limit in bending will give a good indication of the torsional fatigue properties of polished spring wire." But they give no reasons why it should and offer only one sample calculated value, and do not check experimentally the torsion strength of the actual wire whose bending fatigue strength they used in their calculations. The arbitrary use of the ratio $^2/_3$ did not appear to be justified.

Values of stress concentration factor around spherical and cylindrical inclusions have been given by Southwell and Gough (23) and J. N. Goodier (24), and the stresses around an elliptical hole in a plate by C. E. Inglis (25).

Goodier shows that the elastic stress concentration factors at a small spherical cavity in regions of uniform pure shear and of uniform tension stress are 1.91 and 2.72, respectively. These values cannot be used in estimating the effect of such a cavity on fatigue strength in the two cases, since the first is in terms of maximum shear stress, and the second is in terms of maximum tensile stress. From Goodier's results it can be shown that for a cavity in a field of uniform tension stress, the maximum shear stress is increased just 1.91 times. If fatigue failure were known to be determined by maximum shear stress, this would lead to the practical conclusion that the ratio of torsion endurance limit to bending endurance limit is unaffected by the presence of small spherical inclusions which are not too close to the surface.

Since in wire, inclusions are probably lengthened out as a result of drawing, calculations for ellipsoidal cavities would be of interest, but none were found in the literature.[3]

For flaws at the surface of wire, the two-dimensional elastic stress concentrations for circular and for elliptical cylindrical holes might be applied assuming again an arbitrary criterion of fatigue failure. At a minute circular hole in the surface of the wire, for the case of torsion, the maximum shear stress is doubled, and for the case of bending, the maximum shear stress is trebled

[3] A paper by H. Neuber has since been noted, "Der Raümliche Spannungszustand in Umdrehungskerben," *Ingenieur Archiv*, vol. 6, no. 2, April, 1935, p. 133.

(24). Therefore, such a flaw is more effective in reducing the bending fatigue strength than in reducing the torsion fatigue strength.

Still assuming the maximum shear stress criterion, we may consider the elastic stress concentration factors for an elliptical flaw on the surface with major axis placed longitudinally and four times the length of the minor axis. For the case of bending, Inglis' formula gives a factor of 1.5. For the case of torsion, the maximum shear stress is increased by the factor $[1 + (a/2b) + (b/2a)]$, where a and b are the major and minor semiaxes, respectively. This formula is not given by Inglis, but was obtained from his results. The value of the factor in the present instance is 3.12. Hence, it is seen that a longitudinal elliptical surface flaw will reduce the torsion fatigue strength proportionately more than it will the bending fatigue strength. Numerical comparisons are limited by the lack of numerical values for notch sensitivity of the material, but it is apparent that any attempt to calculate the torsion fatigue strength from the bending fatigue strength must take account of the shape of flaws which are permitted by the specifications. A striking example of the formation of four fatigue cracks at the points of stress concentration around an approximately elliptical flaw of a torsion fatigue specimen is shown in Fig. 27.

Different materials under alternating loading react differently to stress concentration.

W. Buchmann (26) concludes that "Notch sensitivity is a utilizable property of the material, determinable by calculation." This is the same quantity which R. E. Peterson (27) calls "stress concentration index" q where

$$q = (k - 1)/(K - 1)$$

where k is the ordinary endurance limit without stress concentration divided by the endurance limit with stress concentration (fillet, hole, etc.), and K is the elastic stress concentration factor and is equal to the maximum stress at a fillet or hole divided by the average nominal stress.

W. Buchmann (26) made fatigue studies in bending and torsion on specimens about 0.3 in. diameter, with circumferential notches of 0.002 in. to 0.08 in. radii. Fatigue strength of annealed materials was practically unaffected by the presence of the notch (q practically equal to zero for very ductile materials), while for hardened steels the reduction in fatigue strength was practically proportional to the calculated increase of stress at the notch (q approximately unity for very brittle materials). For the same material the greatest difference between actual reduction of fatigue strength and reduction calculated from the elastic stress concentration factor occurred with the sharpest notches (q decreases with increased sharpness of the notch in a particular material).

Erich Scheil (28) published a paper on the origin of cracks in steel as a result of heat-treatment, and found a dependence on the internal-stress condition, the separation strength of the steel, and other factors.

The effect of internal stresses on fatigue strength were studied by Buehler and Buchholtz (29), who worked with specimens about 1 in. in diameter. They found that compressive surface stresses of about 30,000 to 50,000 lb per sq in. caused by quenching, raised the bending fatigue limit from 10 to 20 per cent. They quote results which show that large tensile surface stresses reduce the fatigue strength about 15 per cent, and that tensile stresses below 28,000 lb per sq in. have no appreciable effect on fatigue limit. They mention the gradual reduction of internal stresses as a result of fatigue stressing.

The same authors (30) studied the relation between residual temperature stresses and yield point of the material and found higher residual quenching stresses for materials with lower yield point due to the greater amount of plastic deformation which occurred during quenching.

Buehler and Scheil (31) studied the combined effect of temperature and transformation stresses in 2-in. diameter specimens of quenched steels. The nickel content was varied to give various transformation temperatures corresponding to different carbon content in straight-carbon steels. Lower transformation temperatures left tension stresses at the surface. Very high transformation temperatures left compressive stresses. Smaller-diameter specimens had smaller tension or compression surface stresses than large-diameter specimens.

Linicus and Sachs (32), in studying the effect of the die angle and degree of drawing on the internal stresses in wire, measured the internal stress by planing off half the width of the wire and noting the change of curvature. However, this gave only a qualitative measure of stress.

R. M. Brown (33), in measuring residual stresses in cold-drawn mild-steel rods, found that in general, zero stress is at an annulus 0.66 of the diameter, with tensile stress outside and compressive stress within.

General

H. J. French (34) discusses the phenomenon of delayed fractures. Hardened and slightly tempered steels which did not fail at 10 million repetitions, failed suddenly at 30 to 60 million repetitions. The steels were all nickel steels and delayed fractures were observed at hardnesses above the range 45 to 50 on the Rockwell C scale, but not at lower hardnesses. No mention of the same effect in straight-carbon steels was found in the literature.

Another delayed-fracture effect is mentioned by A. Jünger (35) who found that flat bending specimens of hardened chrome-molybdenum steel, in which stress concentration was produced by a small hole, were breaking even after 75 million repetitions, whereas unbored specimens did not break after 5 million repetitions.

2—MATERIALS INVESTIGATED

Three coils of straight-carbon tempered Swedish valve-spring wire were supplied by three different wire manufacturers for the purpose of these tests. The wire was specially selected in regard to freedom from inclusions and for as nearly a perfect surface as could be obtained.

The stress-strain curves for static tensile and torsional tests of these wires are plotted in Figs. 1, 2, and 3, while the dimensions, chemical analyses, Rockwell hardnesses, and strengths are given in Table 1. The ultimate torsion strength was calculated from the formula

$$\frac{5.1 \times \text{maximum twisting moment}}{\text{diameter}^3}$$

Carbon contents for the outer 0.001-in. thick layer were found to be 0.57, 0.59, and 0.59, respectively. Photomicrographs of the boundary material are discussed below.

Fatigue tests were made chiefly on the 0.225-in. and 0.187-in. diameter wires. Complete series of tests were not made on the 0.187-in. and 0.125-in. wires because of lack of time.

Polished longitudinal sections and cross-sections of the 0.187-in. and 0.225-in. diameter wires were examined for inclusions. At 100 magnifications, inclusions were smaller in general than about 0.015 in., and no large inclusions were observed.

The structure at 1000 magnifications of a cross-section of the 0.187-in. wire is shown in Fig. 4, and of a longitudinal section of the 0.225-in. wire in Fig. 5.

The boundary of the 0.187-in. wire at 1000 magnifications is

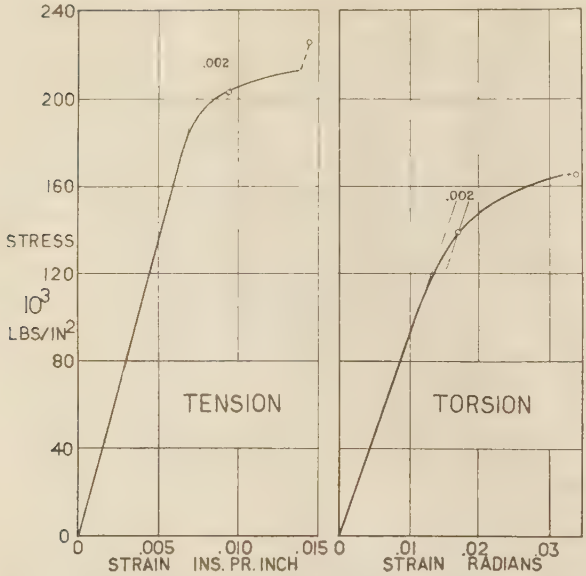

FIG. 1 CURVES FROM STATIC TESTS OF 0.125-IN. DIAMETER TEMPERED SWEDISH WIRE

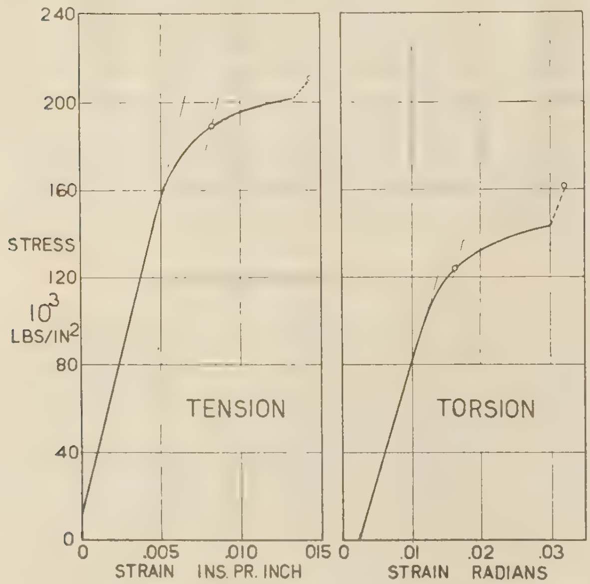

FIG. 2 CURVES FROM STATIC TESTS OF 0.187-IN. DIAMETER TEMPERED SWEDISH WIRE

shown in Fig. 6 in which decarburization to a depth of about 0.0005 in. is shown.

Fig. 7 shows decarburization at the boundary of the 0.225-in. wire, the oxide scale being in focus. Fig. 8 shows another portion of the boundary of the same wire, at which the distribution of ferrite is less disperse.

The photomicrographs show the extent of decarburization better than the carbon analyses previously given, as the latter gave the average carbon content for a layer twice as thick as the actual decarburized layer.

3—DESCRIPTION OF THE FATIGUE-TESTING MACHINES

Bending fatigue tests were made on a short-span rotating-beam-type machine, which was found to work satisfactorily for 0.225-

TABLE 1 RESULTS OF TESTS ON THREE SPECIMENS OF TEMPERED SWEDISH WIRE

Diameter of wire, in.	0.125	0.187	0.225
Coil diameters as received, in.	28	36	36
Chemical analyses:			
Carbon	0.60	0.71	0.65
Manganese	0.35	0.60	0.59
Silicon	0.18	0.21	0.15
Phosphorous	0.023	0.020	0.020
Sulphur	0.026	0.022	0.021
Tensile strength, lb per sq in.:			
Ultimate	226,000	211,000	204,000
Yield point	203,000	190,000	181,000
Torsional strength, lb per sq in.:			
Ultimate	166,100	162,000	166,000
Yield point	139,400	125,000	120,500
Elongation in 8 in., per cent	4.2	5.7	6.2
Elongation in 2 in., per cent	6.9	9.9	10.5
Reduction of area, per cent	51.8	51.7	43.7
Hardness, Rockwell *C*	45.8	42.8	42.2
Stress-strain curve	Fig. 1	Fig. 2	Fig. 3

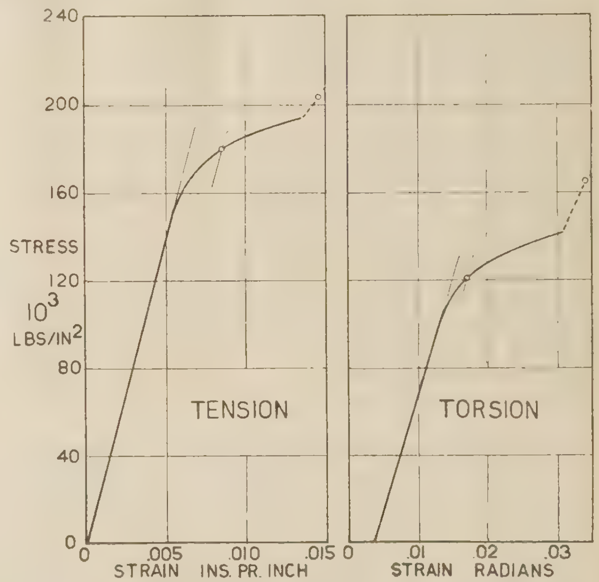

FIG. 3 CURVES FROM STATIC TESTS OF 0.225-IN. DIAMETER TEMPERED SWEDISH WIRE

in. and 0.187-in. diameter wires, but not for 0.125-in. wire. Clear lengths of specimens between $1^1/_2$ in. and $2^1/_2$ in. were used. Grips were cylindrically bored and fitted with an external cone and nut for clamping.

Torsion fatigue tests were conducted on three machines of the constant-strain type shown in Fig. 10. A $^1/_2$-hp driving motor running at 1725 rpm was coupled to a shaft with a variable eccentric. A connecting rod gives an oscillating motion to one of the gripping chucks through a rocker arm of variable radius, the maximum angular motion being about 12 deg.

The second gripping chuck was originally clamped solidly on these machines, but in order to provide a means of determining the stress range at all times while the machine was in operation, a torsion-measuring member was inserted in series with the specimen under test. Two dial indicators measured the relative rotation of the two ends of this member at the extremities of the cycle. This device was calibrated statically under known torsion moments. Inertia effects in it were negligible, the calculated error with the proportions used being 0.1 per cent. Readings of the dials for zero torque were made at the beginning of a test with the specimen in place by unclamping the dead end of the torsionmeter and giving the assembly a slight oscillatory motion to eliminate friction.

A revolution counter was connected to the eccentric shaft. The motor was controlled through a 6-volt relay circuit, a current of 0.125 amp passing through the specimen under test. A

Fig. 4 Structure in a Cross-Section of the 0.187-In. Diameter Wire (×1000)

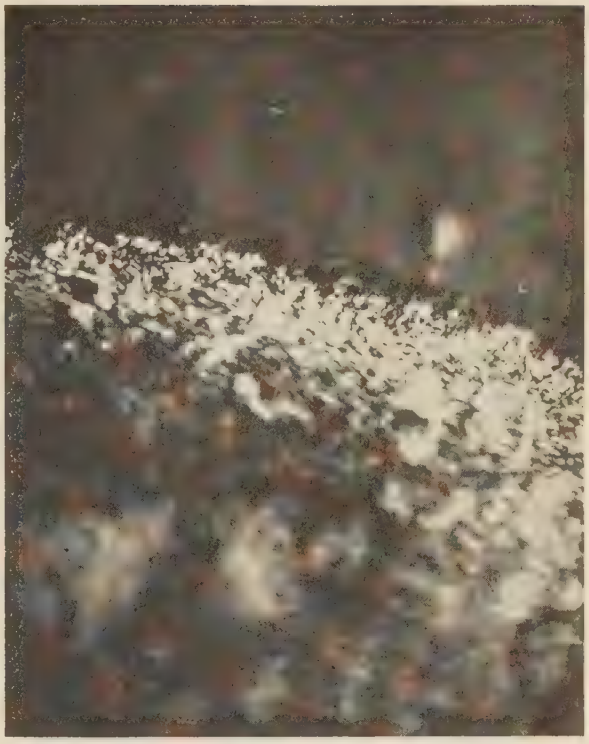

Fig. 6 Decarburization at the Surface of the 0.187-In. Diameter Wire (×1000)

Fig. 5 Structure in a Longitudinal Section of the 0.225-In. Diameter Wire (×1000)

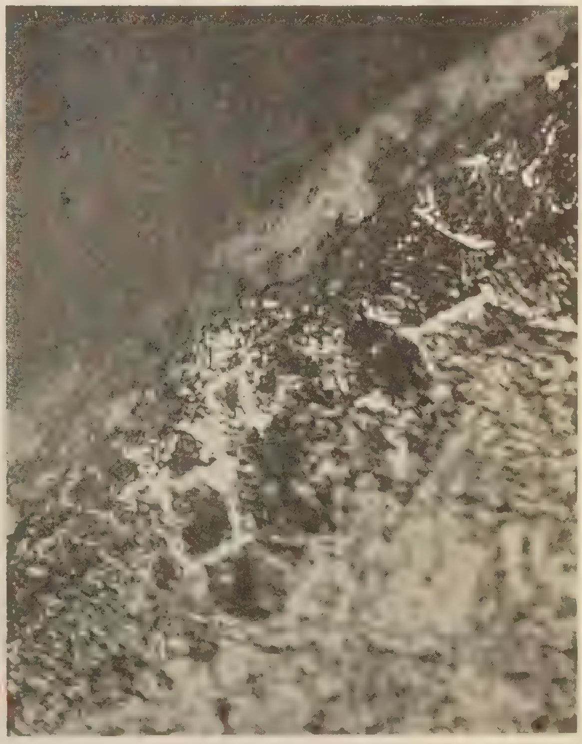

Fig. 7 Decarburization at the Surface of the 0.225-In. Diameter Wire, Showing Also the Oxide Scale (×1000)

spring was provided to shift the oscillating grip a distance of about 0.0625 in., thus opening the circuit. Relays were adjusted so as to remain open after a momentary break in the circuit. This plan did not always stop the motor since contact was often maintained through the two parts of the fractured specimen. A phosphor-bronze leaf-spring contact in series with the specimen was therefore added at the end of the oscillating shaft which positively broke the circuit with a small endwise motion of the shaft and stopped the motor as desired.

Ball-bearings were used throughout, lubrication being maintained by oiling at the end of every one or two million cycles.

Vibration was reduced to a minimum by balancing the rotating eccentric parts and mounting the machines on thick sponge-rubber pads.

For tests at values of mean stress other than zero, a rotational adjustment was provided, which could be varied during operation.

The split chucks were cylindrically bored to 0.5 in. diameter and each was clamped by means of four $^{5}/_{16}$-in. heat-treated bolts.

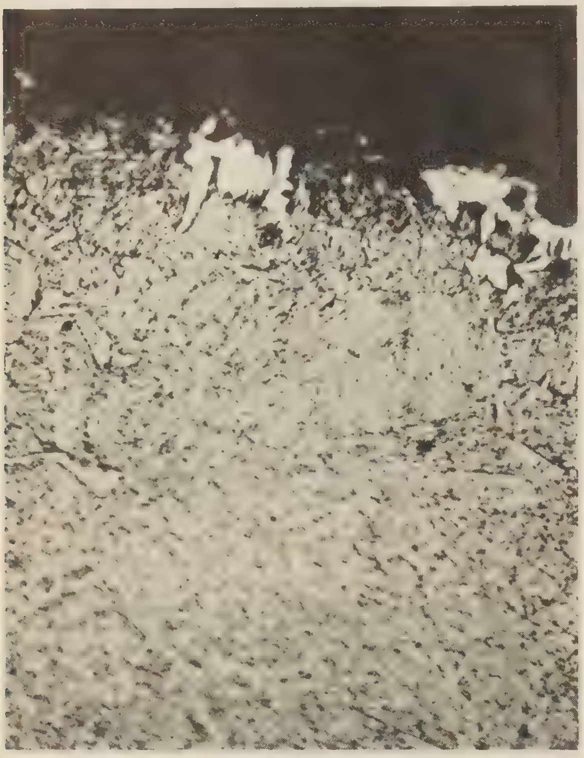

Fig. 8 Another Part of the Same Section of the 0.225 In-Diameter Wire Shown in Fig. 7 Illustrating a Different Type of Decarburization (×1000)

Fig. 9 Seam About 0.001 In. Deep Found in Only One Section of the 0.225-In. Diameter Wire (×100)

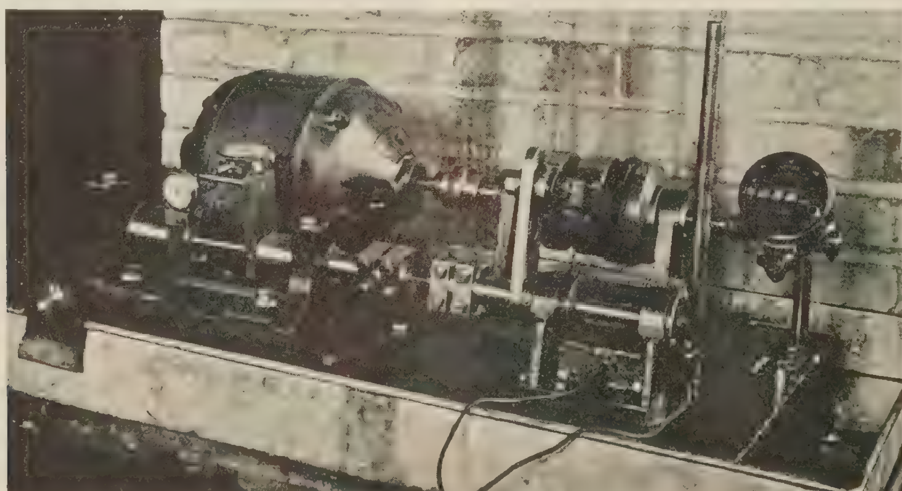

Fig. 10 Torsion Fatigue Machine Showing Motor, Eccentric Shaft, Oscillating Grip, Specimen in Place, Torsion Measuring Dials, and Cut-Out Switch at the Right End of the Oscillating Shaft

Fig. 11 Split Steel Molds; Bending and Torsion Specimens With Cast Type-Metal Ends; Different Types of Steel, Copper and Copper-Lead Bushings, and Specimens Prepared in Different Manners

4—DEVELOPMENT OF GRIPPING METHODS FOR TORSION

Cast-End Method

At the inception of this project no satisfactory methods of gripping were known for the torsion fatigue testing of tempered valve-spring wire in the as-received condition. Failure of straight wire specimens invariably occurred at the grips where the phenomena of stress concentration, microscopic rubbing, oxide formation, and corrosion were all present. The difficulties may be illustrated by Swan, Sutton & Douglas' torsion fatigue tests on Swedish wire (11). Of 25 specimens tested only two failed away from the grips, and another two ran unbroken for ten million stress cycles.

For cold-drawn wires, the difficulties had been overcome by F. C. Lea (2) by the use of cast type-metal ends. Adhesion was obtained by first tinning the ends of the wire which was then gripped in a split steel mold and poured with type metal. Molds and specimens of this type are indicated in Fig. 11. The temperature reached by the central portion of the wire in the process was not above 100 C, and when such high temperatures were reached they were only of momentary duration.

Lea's method was taken as the starting point for the present work with tempered wire. The alloy used was described by Lea (2) as a white metal having a melting point as low as 107 C, and by Goodacre (3) as a type metal melting at 180 C. In the experiments reported in this paper, a matrix metal, a tin-lead solder, and a commercial monotype metal were tried first. The matrix metal contained 28.5 per cent lead, 14.5 per cent tin, 48 per cent bismuth, and 9 per cent antimony, and had a melting point of 105 C. The tin-lead solder had a melting point of approximately 180 C. The monotype metal analyzed 76 per cent lead, 8 per cent tin, and 16 per cent antimony, and had a melting point around 240 C.

The matrix alloy and the tin solder did not adhere to the wire in torsion fatigue trials, whereas the monotype metal always adhered, even for torsional stresses in the wire as high as 140,000 lb per sq in. Monotype metal is relatively inexpensive, and as it seemed to meet the initial requirements it was used in subsequent tests.

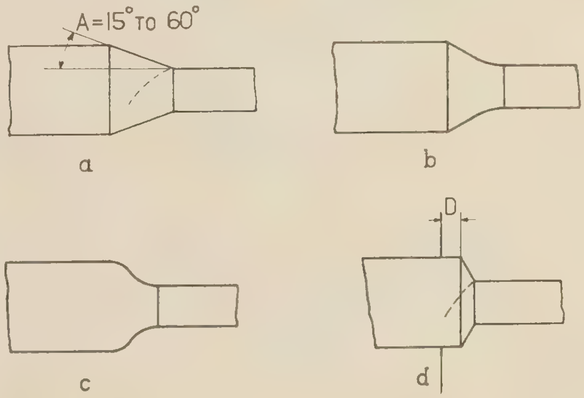

Fig. 12 Shape of the Ends of the Specimens Used in Torsion Fatigue Tests

Specimens of 0.125-in. wire were prepared with cast monotype ends and tested in torsion fatigue. In all cases in the early experiments failure occurred at the inner end of the molded type-metal shank. To reduce stress concentration effects the ends were variously shaped as shown in Figs. 12*a*, 12*b*, and 12*c* and were gripped at varying values of D shown in Fig. 12*d*, but fractures were always at the junction with the white metal thus indicating that stress concentration was probably not the chief cause of failure. For small angles A and for large values of D shown in Figs. 12*a* and 12*d*, respectively, the white metal failed by cracking along a helix. In order to vary the hardness of the white metal, pouring temperatures were varied from 475 C to 300 C, and mold temperatures were reduced to as low as 5 C with no apparent effect on the position of wire fractures.

A few specimens were clamped directly in a split-copper bush

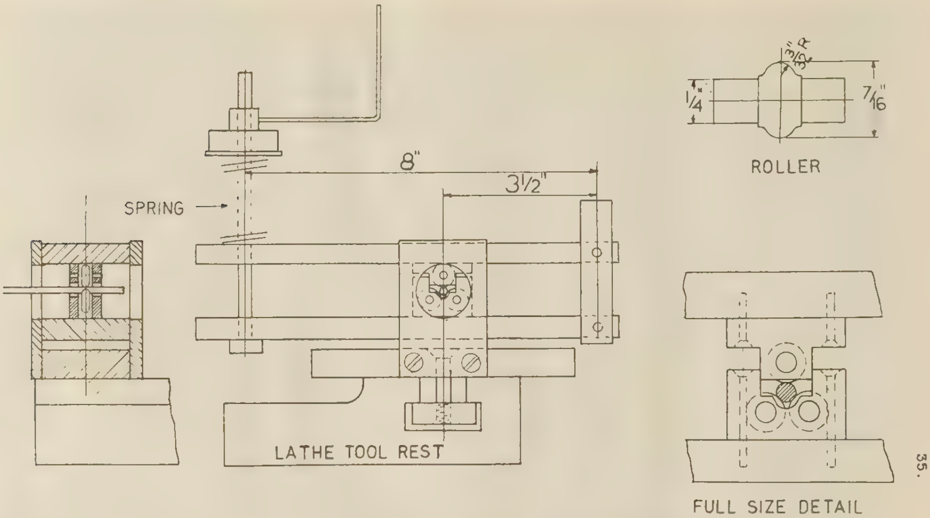

Fig. 13 Details of the Surface-Rolling Device Shown in Fig. 16

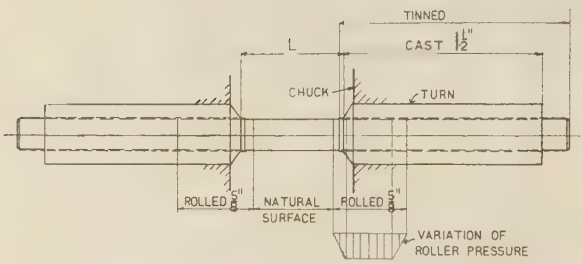

Fig. 14 Dimensions of the Rolled Sections of the Torsion Fatigue Specimens

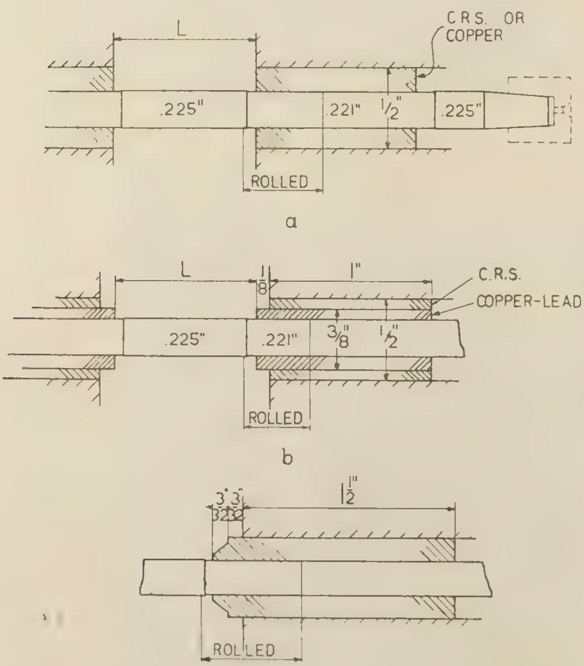

Fig. 15 Bushing Details for Grinding, Rolling, and Gripping Specimens

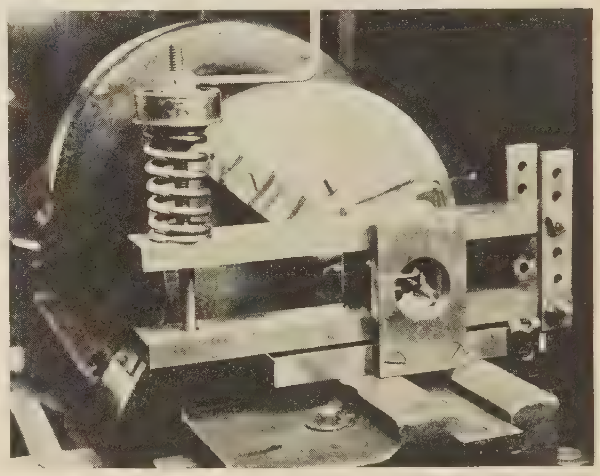

Fig. 16 Device for Rolling the Surface of the Wire. The Spring Was Calibrated to Produce Accurate Rolling Pressures

ing with a 0.009-in. layer of detail paper between the wire and bushing. These failed at the grips.

One specimen was clamped without paper in a split mild-steel bushing and failed at the grip.

Removal of the decarburized layer and polishing is known to greatly increase the fatigue strength of wire. When this was done and followed by tinning and casting on type-metal shanks, failure again, as with other methods, occurred at the grip.

Five specimens of 0.187-in. wire were prepared with cast monotype ends for testing in the rotating-beam machine, in the belief that conditions here would not be so drastic as for torsion and that bending fatigue experiments could be started. Although the different shapes of the ends of the specimens shown in Figs. 12*a*, 12*b*, 12*c*, and 12*d* were tried, failure always occurred at the grips. One bending specimen was not tinned, but cast directly on the wire which had been fluxed in zinc chloride. It failed at the grip, indicating that tinning was not an essential cause of failure.

The presence of a yellowish white oxide at the junction of wire with cast shank suggested a similarity in the gripping problem to that of the failure of axles at a press fit, where stress concentration

and corrosion fatigue due to the formation of oxides are present (8, 9, 10). In the experiments of Thum and Wunderlich remarkable increases of bending fatigue strength were obtained due to surface-rolling. This process was next tried and since it met with success it will be described in detail.

Surface-Rolling of Wire Specimens

The apparatus detailed in Fig. 13 and shown in Fig. 16 was assembled and used to roll the surface of wire specimens at the points where failure had previously been found to occur. The rollers were made of $^7/_{16}$-in. diameter hardened steel with the face crowned to $^3/_{32}$-in. radius. Roller dimensions were taken from the dissertation of O. J. Horger (9). A load of approximately 100 lb was used at the spring, giving roller loads of about 230 lb each. The approximately straight wire specimens were gripped in a lathe chuck and rotated at various speeds. The 0.225-in. wire was rotated at 25 rpm and the 0.187-in. wire at 35 rpm, giving a surface speed of 1.5 fpm. A feed of 0.0055 in. per revolution was used.

With the roller dimensions and loading as given, the maximum shear stress in a perfectly elastic solid calculated by the Herz theory for contact stresses was in the neighborhood of 500,000 lb per sq in. The yield point in shear of the tempered wire is from 120,000 to 140,000 lb per sq in. Consequently, this extremely high stress is not reached but plastic flow takes place to a depth depending upon the roller pressure. The slow rolling speed allows time for plastic flow to occur. As a result of rolling, the surface material is work-hardened, is made more dense, and is left with heavy compressive residual stresses, all of which probably contribute to its increased fatigue strength.

Two portions of the wire specimen, each about $^5/_8$ in. long, were rolled as shown in Fig. 14. The roller pressure was increased linearly from zero to a maximum in a length of $^1/_{16}$ in. to $^1/_8$ in. and similarly reduced to zero at the other end of the rolled portion. The tinning was not permitted to touch the natural surface of the wire in the center portion because of possible penetration and weakening (5). The cast end was kept clear of the end of tinned portion so that perfect adhesion was obtained. The beveled ends of the wire specimen were centered in female lathe centers for turning the type-metal ends to fit the chucks of the torsion machine. It was obviously not necessary for the wire to be straight with this method of gripping.

Before tinning, the ends of the wire were rubbed lightly with emery cloth to remove scale, and fluxed in zinc-chloride solution. They were tinned by dipping into molten 50/50 lead-tin solder at 325 C. The momentary temperature reached at a point in the wire just clear of the tinning did not in general exceed 75 C. This was measured by inserting a thermocouple in small holes drilled in pieces of $^1/_8$-in. and $^1/_4$-in. drill rod, and tinning these as for the wire specimens.

The monotype metal was poured at 325 C and the molds were usually at or slightly below room temperature. The split molds weighed 0.60 and 1.25 lb for the 0.187-in. and 0.225-in. wires, respectively. The metal cooled quickly and wire temperatures as measured by thermocouples did not exceed 75 C.

The combination of cast ends with surface rolling of the wire was successful for alternating torsion fatigue tests on as-received wire and on wire with the surface ground. Practically every specimen failed away from the gripping points. The method was not successful for shot-blasted wire, since the surface material of this wire is strengthened greatly by work-hardening. The method was successful for bending fatigue tests in a short-span rotating-beam machine on wire in the as-received, ground, shot-blasted, and shot-blasted-and-blued conditions.

To obtain torsion fatigue results for shot-blasted wire, a surface layer of approximately 0.0015 in. was removed from the two end portions of the specimens after which they were polished and rolled. Since the wire specimens were not perfectly straight it was necessary to rotate them in a lathe and clean them with emery cloth. All scratches had to be removed to eliminate the danger of the specimen breaking at the scratches. The few specimens prepared in this way did not fracture as desired, probably due to scratches from the emery, but the double strengthening due to removal of decarburized layer and to surface rolling should more than counterbalance the strengthening in the center portion of specimen due to shot-blasting. Further work with this method should yield results.

When the cast-end type of specimen was tested in a zero-to-maximum torsion-stress cycle, there was a continuous angular creep in the type-metal ends. The effect of this creep was to shift the stress range to a position with a lower mean stress value so that it was necessary to adjust the machines at frequent intervals in order to maintain the intended stress cycle. The cast-end method was consequently abandoned for zero-to-maximum stress cycles, although it is probable that an automatic method of maintaining the proper stress cycle can be worked out. This may be a desirable development as the cast-end method has the great advantage of reducing stress concentration to a minimum.

Split-Bushing Methods

For tests in a zero-to-maximum torsion-stress cycle, split bushings were used to grip the wire. In conjunction with the bushings surface rolling was always used to strengthen the wire in the critical region. Bushing materials experimented with were ordinary cold-rolled steel, hard-drawn copper, and a copper-lead alloy with about 25 per cent lead. Steel was used because of its high coefficient of friction, requiring lower gripping pressures, while copper-lead was used because of its successful application in the pulsating tensile tests of Shelton and Swanger (13). The friction coefficient of brass was too low for satisfactory gripping. In some cases the ends of bushings were faced square with no attempt at relieving stress concentration, while in other cases they were beveled in an attempt to reduce the pressure peak occurring at the ends. When set in the machine they were chucked sometimes flush at the ends, and sometimes protruding a short distance, which was believed to reduce the pressure peak due to gripping. In some cases a thickness of paper was used between the wire surface and the bushing bore. Rolling and subsequent gripping of the wire were done either directly on the natural surface or on a portion which was polished after being reduced about 0.004 in. in diameter to remove the decarburized layer.

The wire diameter was reduced either by grinding between centers which could only be done on samples specially selected for straightness, or more practically by hand in a lathe using strips of successively finer emery cloth. Bushings were drilled solid with a twist drill and sometimes finish-bored by a gun reamer with which it was attempted to bore them 0.0005 in. smaller than the neat-wire size. They were split into halves by a $^1/_{64}$-in. milling cutter, and in one case each half was slotted to provide greater flexibility in clamping around the surface of the wire.

Five specimens of 0.225 in. wire in the as-received condition were selected for straightness and ground between centers to 0.221 in. diameter using the tapered-end fittings shown in Fig. 15*a*. They were rolled and gripped by the methods of either Fig. 15*a* or Fig. 15*b*. Specimens Nos. 350 and 351, shown in Fig. 25, were gripped in the copper-lead bushing detailed in Fig. 15*b*. Both of these specimens fractured in the center as shown in Fig. 25. However, the fracture of specimen No. 350 was at a surface flaw as is evidenced from the stress-repetition or S-N curve in Fig. 18. Specimen No. 352 was gripped in a copper

bushing 1 in. long, as shown in Fig. 15*a*, and was run 10.1 million cycles of 0 to 97,000 lb per sq in. stress without breaking. Specimen No. 353 shown in Fig. 25 was gripped in a cold-rolled steel bushing 1 in. long and broke in the center portion. Specimen No. 354 similarly gripped in a steel bushing was removed unbroken after 4.4 million cycles. Very little galling of the wire surface was evidenced. Copper-lead bushings could not be used more than once because of plastic deformation. The method may be considered successful but somewhat elaborate.

In an effort to simplify the procedure, the reduction in diameter of the gripped portions of specimens was omitted. The natural surface of the wire, or 0.187 in. diameter in this case, was rolled as required, the scale was removed with fine emery paper, and the wire was gripped in split copper bushings. Although the wire had been straightened by hand bending none of the specimens were actually straight so that gripping in the torsion machine would introduce small static bending stresses which were neglected. Specimens Nos. 201 to 207 were prepared in the manner described and gripped in copper bushings having an end detail as shown in Fig. 15*c*. Specimens Nos. 201, 205, and 206 fractured clear of the grips while specimens Nos. 202, 203, and 204 broke at the grips. Specimen No. 207 ran 10.1 million cycles at 0 to 105,000 lb per sq in. stress. There was noticeable galling and probably penetration of the copper into the steel wire at points near the inner ends of the bushings where the pressure was greatest. Fatigue cracks started at such points. It was believed that the decarburized surface was particularly susceptible to this welding or seizing action and the next group of specimens Nos. 208 to 210, were prepared with the gripped portions reduced in a lathe to 0.183 in. by emery cloth previous to the surface-rolling operation. Fig. 15*c* indicates the tapered end of bushing and projection from the chuck as used for all three specimens. Specimen No. 208 was gripped in a steel bushing and broke at the grip. Specimens Nos. 209 and 210 were gripped in split copper bushings but were first wrapped with a 0.003-in. layer of a high-grade linen bond paper. The bushings had been bored to 0.185 in. diameter and therefore had to be forced onto the paper-covered specimen. Both specimens broke clear of the bushings. The paper was somewhat disintegrated at the inner ends of bushings, as is usual, due to small relative motion. There was also some fusing of copper to steel.

It is believed that with better distribution of gripping pressure most of the failures at the grips in these experiments might have been avoided. A good bond-paper filler will compensate for irregularities in the surface of the bushing bore, but it is also necessary that the gripping pressure be distributed radially around the wire surface rather than along two lines which are diametrically opposite to one another. To obtain this distribution, an undersized bushing bore was tried as mentioned, and in some cases a paper filler was placed between the outside of the bushing and the bored hole of the torsion-machine chucks. This had the effect of forcing-in the opposite edges of the bushing halves and thus improving the distribution of pressure.

Welding of the bushing material to the wire surface had troublesome consequences in the tests. Apart from providing points of weakness, it reduced the effective length of specimen between grips and thus, since the torsion machines are constant-strain machines, increased the range of stress applied to the specimen. The action was progressive and called for frequent strain adjustments to keep the stress range within reasonable limits. This effect was observed on the 0.187-in. wire gripped on the natural surface and when reduced to 0.183 in. by emery cloth and paper. It was also observed on the 0.225-in. wire ground between centers to a 0.221-in. diameter gripping seat when gripped in copper-lead bushings (specimen No. 351), but was not observed when gripped in steel or copper bushings (specimens Nos. 354 and 352). These

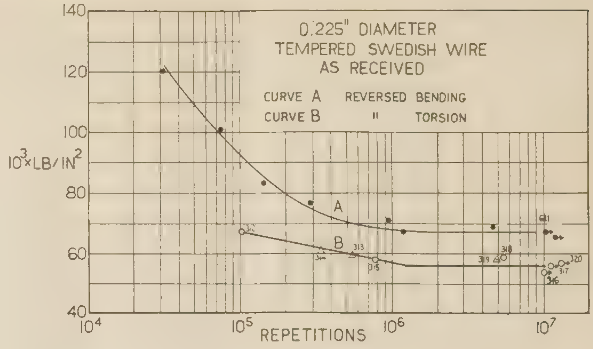

Fig. 17

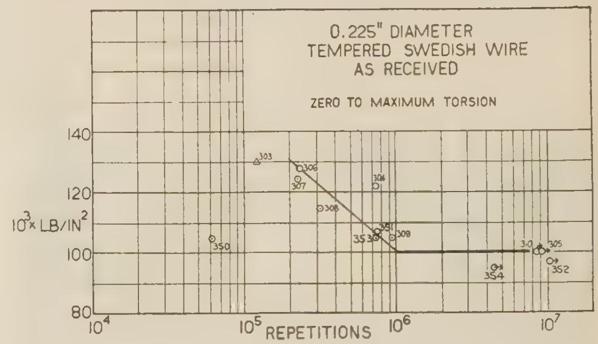

Fig. 18

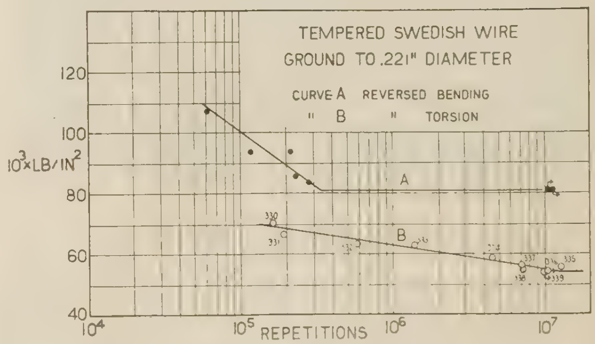

Fig. 19

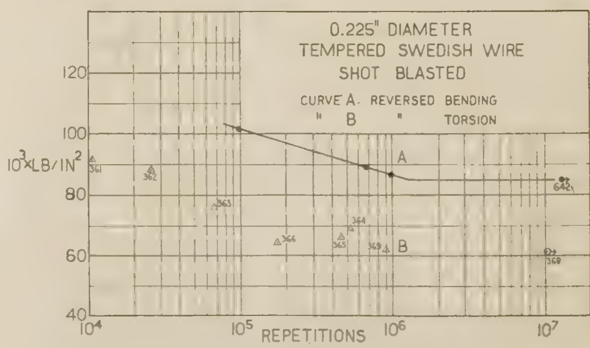

Fig. 20

TABLE 2 VARIATIONS IN STRESS RANGE IN SEVERAL OF THE TESTS

Specimen no.	Type grip	Variation in range
207	Copper bushing	Increase 6% in 2 million cycles
208	Steel bushing	Increase 3 to 5% in 0.2 million cycles
224	Monotype metal	± 7% in 3.6 million cycles
226	Monotype metal	Decrease 1.5% in 2.5 million cycles
364	Monotype metal	Decrease 8% in 0.5 million cycles
354	Steel bushing	Decrease 1% in 4.5 million cycles
351	Copper-lead metal	Increase 7.5% in 0.7 million cycles
352	Copper bushing	Decrease 4.5% in 2.2 million cycles
338	Copper bushing	Decrease 1.0% every 2.5 million cycles

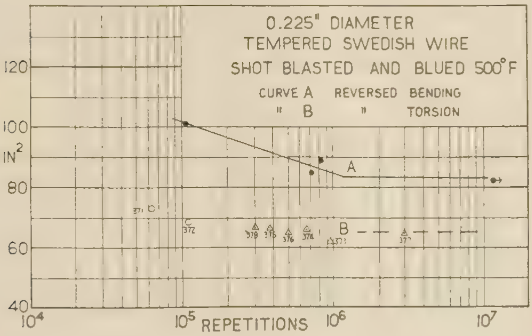

FIG. 21

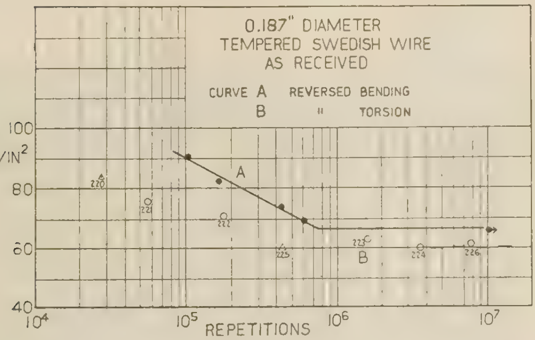

FIG. 22

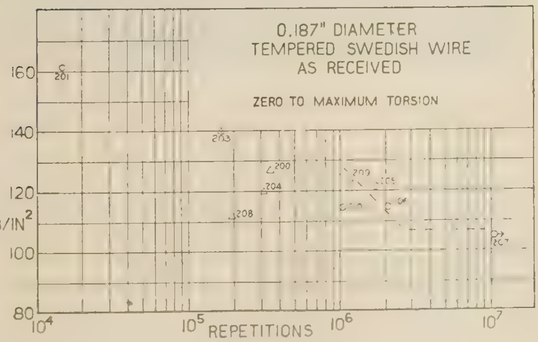

FIG. 23

latter results do not agree with Shelton and Swanger's (13) conclusion that copper-lead causes less galling of the surface than copper, and it may be, as mentioned previously, that the distribution of gripping pressure was the determining factor.

The stress range did not remain constant throughout the endurance test of any one specimen, although in some instances the variation was very small. In contrast with the phenomenon mentioned previously, some specimens gripped in split bushings showed a gradual reduction in stress range, probably due to a loosening of the wire at the inner ends of the bushings. The specimens with cast type-metal ends always showed a reduction in range which seemed to be due to relaxation under the gripping pressure, and consequent loosening in the chuck. This was corrected easily by tightening the gripping bolts. The effect was greatest at high stress ranges when specimens heated up considerably, but was quite small in tests near the fatigue limit. Regular attention and readjustment was thus necessary in all torsion tests to maintain stress ranges within satisfactory limits. The observed variations in stress range for some individual specimens are given in Table 2.

5—TEST PROCEDURE

The wire surface was examined by etching in a solution of equal amounts of concentrated hydrochloric acid and water maintained at 80 C. The wire was reduced in diameter in steps of about 0.0005 in. and examined at each stage until a reduction of about 0.004 in. had been made. The surface of all three sizes was exceptionally homogeneous and free from defects. With much deeper etching longitudinal streaks appeared which were probably not related to the original surface.

A seam about 0.001 in. deep, found in a photomicrograph of the 0.225-in. wire, shown in Fig. 9, could not be located in other sections and therefore probably did not extend a great distance in the wire.

Fractures were examined directly and at about 10 magnifications. In general, inclusions, if present at the point of origin of the fatigue crack, were too small to be seen, but their presence was often indicated by a slight distortion of the fractured surface at this region, as it is known that the value of stress and its direction in the region of a cavity are largely determined by the shape of the cavity. Some individual specimens which did not show up well in the fatigue tests showed failure to have begun at an inclusion or seam as, for example, specimens Nos. 350 and 221, shown in Figs. 25 and 27, respectively. A few cross-sections of failed specimens were polished and examined at 100 magnifications for cracks without success.

The clear length between grips of torsion specimens was determined by the maximum angular travel of the oscillating chuck, and the stress range desired for the individual specimen. Values of length L used in the tests were approximately as follows:

Wire diameter, in.	L
0.125	$^5/_8 \pm ^1/_8$
0.187	$^3/_4 \pm ^1/_8$
0.225	$^7/_8 \pm ^1/_4$

Specimens were cut to a length of about $4^1/_2$ in. and prepared with either cast ends or split bushings as previously described.

In the early tests on 0.225 in. wire, it was used just as it came from the coil, having a slight natural curvature. In all later tests, for convenience, the wire was hand-straightened by being bent in opposition to this natural curvature. Residual stresses were no doubt present in both cases.

Care was taken not to scratch the wire in preparing specimens, as it was found that scratches were often, though not always, the points at which the deciding fatigue crack started.

Specimens were mounted in the testing machine which was started at approximately the desired stress range. By adjustment of the angular travel of the oscillating grip the desired stress range would be obtained in a few thousand stress cycles. Adjustments were made as required to maintain a fairly constant range.

In general, a point on the fatigue diagrams of Figs. 17 to 23 represent a stress cycle which did not vary more than ±5 per

cent. Error in reading torque was about 0.5 per cent, and the error in initial calibration is estimated at 0.5 per cent.

Torsion stresses were calculated by the formula

$$\text{Maximum shear stress} = \frac{5.1 \times (\text{maximum twisting moment})}{\text{Diameter}^3}$$

Bending stresses were calculated by the formula

$$\text{Maximum bending stress} = \frac{10.2 \times (\text{maximum bending moment})}{\text{Diameter}^3}$$

Because of its possible effect on the relative strength in torsion

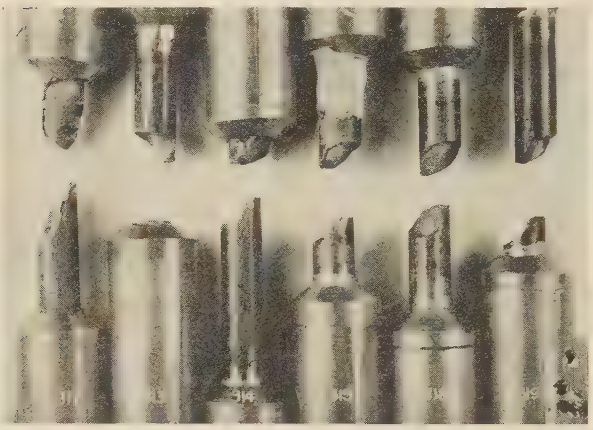

Fig. 24 Typical Fatigue Fractures of 0.225-In. Wire Tested in Alternating Torsion. Wire Used in As-Received Condition

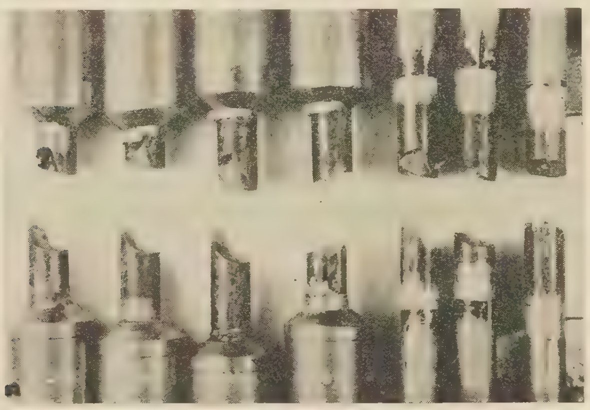

Fig. 25 Typical Fatigue Fractures of 0.225-In. Wire Tested in Zero-to-Maximum Torsion. Wire Used in As-Received Condition

and bending fatigue, the presence of residual stress in the wire was verified experimentally. The method neglects any circumferential stress which may be present and, therefore, at best is only approximate. A flat surface about $^1/_{32}$ in. wide was ground on one side of a length of wire and the change in curvature of the wire noted. From the change in curvature and the calculated segmented area of the portion removed the longitudinal stress in the surface material can be found. A change of curvature in one direction indicates a tension stress, while a change in the opposite direction indicates a compression stress.

Measurements were made on straight pieces of wire in different conditions and on half-coils of wire specially prepared for the measurement of coiling stresses. Fig. 30 shows the device for measuring the coiling stresses with a half coil clamped in position and an ordinary micrometer arranged to measure deflections. Contact was determined by the use of ear phones and a battery circuit.

Where residual stress values are given, the possible error might easily be ±50 per cent although the figures given are averages of 2 to 4 separate determinations which usually agreed within ±25 per cent. Measurements of coiling stresses were made in a limited time and only relative values of stresses are available.

6—RESULTS OF TESTS

The results of tests on 0.225-in. and 0.187-in. wire are plotted as stress against the logarithm of the number of repetitions to failures on the S-N diagrams of Figs. 17 to 23. Fractures which occurred away from the ends are indicated by circles while specimens which did not fracture are indicated by a circle and an arrow. In some cases, since they provide useful information, fractures which occurred at or near the grips are included, and are indicated by triangles.

Results of Tests on 0.125-In. Wire as Received

A complete series of tests was not made on 0.125-in. wire as received, but the results of tests on two specimens give some idea

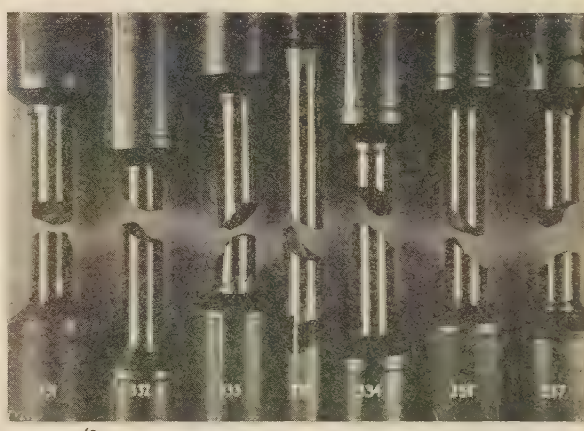

Fig. 26 Typical Fatigue Fractures of 0.221-In. Centerless-Ground Wire Tested in Alternating Torsion

of the fatigue strength in a zero-to-maximum torsion stress cycle. One specimen without failure withstood 15 million repetitions of a stress cycle which at the beginning of the test was from zero to 125,000 lb per sq in., at 10 million zero to 107,000 lb per sq in., and at 15 million zero to 106,000 lb per sq in.

Another specimen broke after 5.7 million repetitions of a stress cycle which at the beginning of test was from zero to 124,000 lb per sq in., at 0.26 million cycles it was from zero to 116,000 lb per sq in., and at the end of the test it was from zero to 113,000 lb per sq in.

From these two tests the zero-to-maximum torsion endurance limit for 0.125 in. wire as received may be estimated at something over 110,000 lb per sq in.

Results of Tests on 0.225-in. Wire as Received

The bending fatigue results show little scatter, as seen from the curves in Fig. 17, and give a smooth curve, with the endurance limit at 67,000 lb per sq in. which is 33 per cent of the ultimate

strength of 204,000 lb per sq in. Reversed torsion tests show an endurance limit of 56,000 lb per sq in. which is 34 per cent of the ultimate torsional strength of 166,000 lb per sq in. Bending fractures shown in Fig. 29*a* were smooth and showed clean material. Torsion fractures also were all in homogeneous material, the crack starting at about 45 deg to the axis forming two roughly symmetrical fractured surfaces as shown in Fig. 24. Reversed torsion endurance limit is 0.84 times the reversed bending endurance limit.

Zero-to-maximum torsion tests showed an endurance limit of 100,000 lb per sq in. as observed from Fig. 18. Fractures of this specimen, shown in Fig. 25 were unsymmetrical and of a different type. The fatigue crack started at about 45 deg and curved around the specimen in circumferential and longitudinal directions. Specimen No. 350 gave a point well off the curve and the position of the surface flaw responsible for failure can be seen in Fig. 25. Fractures otherwise showed clean material.

Tests of 0.225-In. Wire Reduced to 0.221 In. in a Centerless Grinder

This series of tests was run to determine the effect of a surface-grinding treatment which might be commercially practicable. A single pass across a 60-grit wheel was used. The surface though smooth in appearance was considerably coarser than a surface polished with No. 1 emery paper. It was found that due to slight curvature of the wire before grinding, a variable thickness of layer was removed. Only specimens which had "cleaned up" all over were tested.

Fig. 27 Typical Fatigue Fractures of 0.187-In. Wire Tested in Alternating Torsion. Wire Used in As-Received Condition

Fatigue curves for this test are shown in Fig. 19. The bending endurance limit is 81,000 lb per sq in., an increase of 21 per cent over the value for wire as received, in spite of the fact that grinding marks are circumferential. Bending fractures of the specimens used in this test, shown in Fig. 29*b*, are seen to be slightly stepped and ragged, and are not as clean cut as for the as-received wire.

The endurance limit in reversed torsion from the curve in Fig. 19 is 54,000 lb per sq in., which is 4 per cent lower than the value for as-received wire and 0.67 times the reversed bending endurance limit of ground wire. Fractures occur at $4^1/_2$, 6, and 13 million cycles, and the curve does not become horizontal until about 10 million cycles are reached.

The "delayed fractures" of French (34) and Jünger (35) are suggested by these results. French found that fractures occurred after 10 million cycles in nickel steels harder than about 45–50 Rockwell *C*, but not in steels with lower Rockwell values. Removal of the soft outer surface may leave a surface which is in the hardness range where delayed fractures occur, and the usual test to 10 million cycles is not sufficient. Also the grinding marks might provide stress concentrations and cause delayed fractures such as Jünger observed. These effects are not present in the bending fatigue curve, however.

An examination of the fractured torsion specimens showed that a cross-section of specimen No. 330 revealed no cracks or seams. The fractured surface was stepped and a crack extended at about 45 deg into the metal, suggesting the presence of an inclusion. Specimens Nos. 332, 333, 334, 335, and 337, Fig. 26, showed slight irregularities on the fractured surface at the region of crack formation which might have been flaws, whose effect was accentuated in some manner by the grinding.

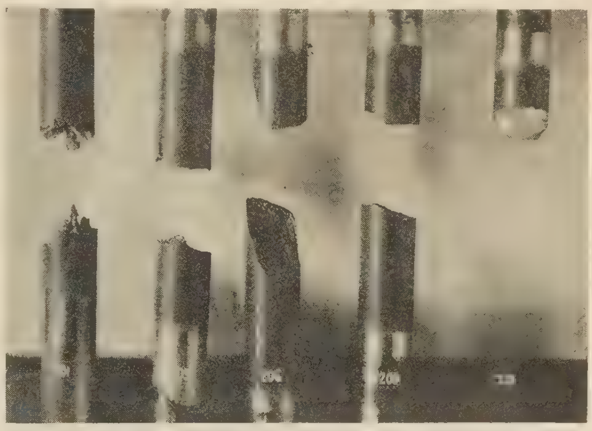

Fig. 28 Fatigue Fractures of 0.187-In. Wire Tested in Zero-to-Maximum Torsion. Wire Used in As-Received Condition

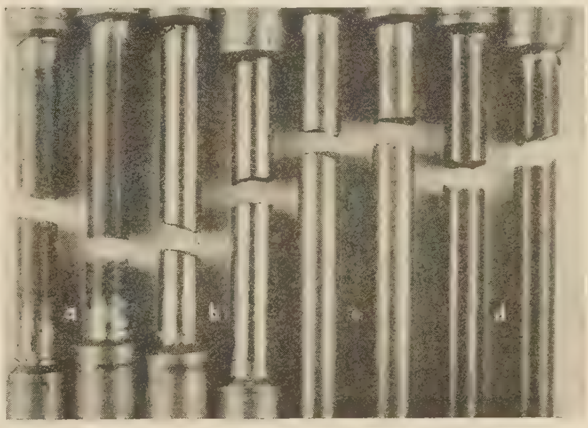

Fig. 29 Typical Bending Fatigue Fractures of 0.225-In. Wire. (*a*) As-Received Condition; (*b*) Ground to 0.221 In. Diameter; (*c*) Shot-Blasted Condition; (*d*) Shot-Blasted and Blued Condition

Results of Tests on 0.225-In. Shot-Blasted Wire

Wire in the as-received condition was shot-blasted in a standard revolving drum for 25 min with air at a pressure between 70–75 lb per sq in.

The reversed bending endurance limit as shown by the curve in Fig. 20 was 85,000 lb per sq in. which is 27 per cent higher than for wire as received. Bending fractures of the specimens are shown in Fig. 29*c*.

TABLE 3 BENDING AND TORSION ENDURANCE VALUES FOR TEMPERED SWEDISH VALVE-SPRING WIRE[b]

Diameter, in.	0.125[b]	0.225[b]	0.221[c]	0.225[d]	0.225[e]	0.187[a]
Reversed bending endurance limit, lb per sq in.	......	67,000	81,000	85,000	83,000	67,000
Per cent of ultimate tensile strength	......	32.8	......	......	......	31.8
Reversed torsional endurance limit, lb per sq in.	......	56,000	54,000	61,500+	65,000+	60,000
Per cent of ultimate torsional strength	......	33.7	......	......	......	37.0
Ratio of reversed torsional stress to reversed bending stress	......	0.84	0.67	0.72+	0.78+	0.90
Zero-to-max. torsional stress endurance limit, lb per sq in.	111,000+	100,000	......	......	......	107,000
Ratio of zero-to-max. torsional endurance limit to reversed bending stress	......	1.49	......	......	......	1.60
Ultimate tensile strength, lb per sq in.	226,000	204,000	......	......	......	211,000
Ultimate torsional strength, lb per sq in.	166,100	166,000	......	......	......	162,000

[a] Based upon 10 million stress reversals.
[b] Wire tested in as-received condition.
[c] Wire ground to size from 0.225 in. diameter with No. 60 grit.
[d] Wire shot-blasted before testing.
[e] Wire shot-blasted and blued before testing.

All torsion fractures occurred at or near the grips and these are shown on the diagram in Fig. 20 by triangles. One specimen ran 10 million cycles at 61,500 lb per sq in. without fracture. The true fatigue curve will lie above the triangular points in Fig. 20 representing fracture at the grips so that the endurance limit is estimated to be an unknown amount above 61,500 lb per sq in.

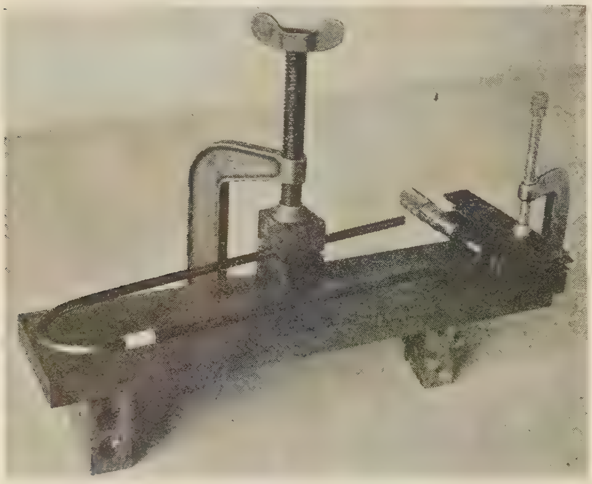

FIG. 30 DEVICE FOR MEASURING COILING STRESSES IN A HALF-COIL
(A flat spot is ground on the inner surface of the coil and the deflection of the arm is measured by a micrometer. Earphones and a battery circuit are used to record contact at the micrometer.)

RESULTS OF TESTS ON 0.225-IN. SHOT-BLASTED-AND-BLUED WIRE

After shot-blasting the specimens were blued at 500 F for 30 minutes in a Homo furnace and cooled in the furnace.

The bending-endurance limit shown by the curve in Fig. 21 is 83,000 lb per sq in. which is about 24 per cent greater than for wire as received. Fractures were slightly irregular but clean, and are shown by the specimens in Fig. 29*d*.

Only two reversed-torsion specimens fractured clear of the grips although some of the other fractures started clear of the grips and extended to them. None of the specimens ran 10 million cycles without fracture, but the endurance limit is probably above 65,000 lb per sq in.

RESULTS OF TESTS ON 0.187-IN. WIRE AS RECEIVED

The bending fatigue limit for 0.187-in. wire as-received is shown on the curve in Fig. 22 to be 67,000 lb per sq in., which is about 32 per cent of the ultimate strength of 211,000 lb per sq in.

Seven points are shown for the reversed-torsion endurance curve, two of which represent failure at the grips. No specimen ran through 10 million cycles without failure, up to the time of making the diagram, but the endurance limit is very probably at 60,000 lb per sq in., which is 0.90 times the reversed bending endurance limit, and 37 per cent of the ultimate torsional strength.

Some torsion fractures of the 0.187-in. specimens are shown in Fig. 27. Specimen No. 221 failed at a longitudinal surface flaw or inclusion about 0.04 in. long. Two diagonal cracks are seen extending from the points of stress concentration at each end of the flaw.

The curve for zero-to-maximum torsion tests on 0.187 in. wire is shown in Fig. 23. One specimen was unbroken after 10 million reversals at 105,000 lb per sq in. A greater number of significant points was not obtained because of difficulties encountered with seizing of the copper-bushing to the wire thereby increasing the stress range in an erratic manner. The endurance limit is estimated at 107,000 lb per sq in. Specimen No. 210 failed clear of the grips but a radial crack about 0.03 in. long could be seen on the fractured surface, which explains the position on the diagram. Specimen No. 201 was accidentally run at a very high stress. Fracture started at many points forming a ragged cone and a conical hollow. There was considerable slipping in the copper bushing at this high stress.

A tabulation of all fatigue results is given in Table 3.

RESIDUAL STRESSES

The following results of residual stress measurement are included for their suggestive value. Errors of ±50 per cent or more may be present.

(*a*) *As Received.* One piece of 0.187-in. wire and one piece of 0.225-in. wire, just as they came off the 36-in. diameter coils with a slight permanent set or curvature, were used to determine the residual stress along four edges spaced 90 deg around the wire. No check measurements were made in this case. The results of this test are given in Table 4.

TABLE 4 LONGITUDINAL RESIDUAL STRESSES IN WIRE AS RECEIVED[a]

Diameter of wire, in.	0.187	0.225
Kind and location of stress:		
Inside edge of coil, tension	180,000	90,000
Outside edge of coil, compression	70,000	120,000
Flat side (1), tension	40,000	30,000
Flat side (2), tension	15,000	20,000

[a] All values in lb per sq in.

The results in Table 4 show very well that even coiling to 36 in. diameter leaves large tensile stresses at the inside of the coil and corresponding compression stresses at the outside. The averages of the values measured along the two flat sides are approximately 25,000 lb per sq in. in both cases, which is probably not sufficient to materially affect fatigue strength (29).

(*b*) *Ground 0.221-In. Diameter Wire.* Measurements on three samples of this wire were erratic and gave compressive longitudinal stresses from 0 to 8000 lb per sq in. The fact that an uneven thickness of layer was removed will explain the variation in stress measurements.

(*c*) *Shot-Blasted 0.225-In. Diameter Wire.* Measurements on three samples gave values of longitudinal compressive stress which were within ±12 per cent of 165,000 lb per sq in.

(*d*) *Shot-Blasted and Blued 0.225-In. Diameter Wire.* Three measurements gave values of longitudinal compression within ±14 per cent of 27,600 lb per sq in. The bluing treatment reduced shot-blasting stresses to one-sixth their value.

(*e*) *Measurements on Fatigue Specimens.* Bending specimen No. 611, made of 0.225-in. diameter as-received wire, after 10 million reversals at 67,000 lb per sq in., showed an average value of longitudinal tension stress, for four sides of the wire of 73,000 lb per sq in. No other specimens were available to check this result which would indicate that residual tension stresses from tempering do not disappear under fatigue loading, and may, therefore, have an effect on the fatigue strength.

Bending specimen No. 642 of shot-blasted wire had run 13 million reversals at 85,000 lb per sq in. The only two measurements obtained on this specimen were zero and 56,000 lb per sq in. longitudinal compressive stress. No other specimens were available for check measurements. It would appear that the heavy compressive stress due to shot-blasting does not remain under fatigue stressing and consequently tests to 20 or 30 million cycles might give lower fatigue values than those obtained.

(*f*) *Coiling Stresses.* After coiling as-received wire in a 2-in. diameter coil, heavy residual longitudinal tensile and compressive stresses were present on the inside and outside of the coil, respectively. Bluing reduced these stresses to about one-third their original value, but they were still appreciable in magnitude.

7—CONCLUSION

The zero-to-maximum torsion endurance limits of 110,000, 107,000, and 100,000 lb per sq in. for as-received wire are about 70 per cent higher than the values of 68,000 and 58,000 lb per sq in. estimated from the results of experiments on completed springs of similar material by F. P. Zimmerli (12). Residual tension stresses at the inside of the coils, even if they were shown to be present after the bluing treatments, would not be sufficient to explain such large differences in fatigue strengths, as Buehler and Buchholtz' results (29) show variations of only 15 or 20 per cent due to internal stress.

The results given in this paper may be considered as ideal values as it is possible to attain if no serious flaws are present. The twelve 0.225-in. specimens tested in zero-to-maximum torsion, the results of which are plotted in Fig. 18, represent a total length of not over 12 in. of wire actually submitted to the repeated stressing. In a single complete spring, possibly 40 in. of wire would be under test and if there were only one flaw in each spring such as that in specimen No. 350, indicated in Fig. 18, it is obvious that lower fatigue values would be obtained in the tests on complete springs. The only remedy for such a condition would be the elimination of all serious flaws in the material.

Comparison may also be made with Swan, Sutton, and Douglas torsion results (11) on straight specimens of 0.182-in. diameter tempered Swedish wire with natural surface. The present zero-to-maximum value of 110,000 lb per sq in. for 0.125 in. wire is about 70 per cent higher than the results of their tests which were made at a ratio of minimum-to-maximum stress of 1:4.

The consistently high values of torsion endurance limit obtained for three sizes of wire in the experiments reported in this paper might suggest the presence of some consistent error in the work. However, the tests were made on three machines which were separately calibrated statically; calibration factors agreed closely and test results for the same wire on different machines were consistent among themselves.

Although ratios of reversed-torsion fatigue strength and of zero-to-maximum torsion fatigue strength to reversed-bending fatigue strength have been obtained for 0.187 in. and 0.225 in. wires and given in Table 3, it has not been possible to correlate these ratios with the factors which largely determine them, such as the shape of inclusions and of surface flaws. The extent and relative effect on bending and torsion fatigue strengths of anisotropy due to the drawing process has not been investigated. The effect of residual stress, if present, on the ratio of bending and torsion fatigue strengths would appear to call for investigation. Until information is available in the directions indicated it is not apparent that the bending fatigue strength of a wire will be a certain measure of the torsion fatigue strength of that wire.

Acknowledgment

The tests reported in this paper were sponsored by the A.S.M.E. Special Research Committee on Mechanical Springs, and were carried out through the Engineering Research Department of the University of Michigan, Ann Arbor. Acknowledgment is here made to members of the subcommittee, in particular to F. P. Zimmerli, to Profs. A. E. White and E. L. Eriksen for assistance of various kind, and to Prof. S. Timoshenko for helpful criticism and suggestions. Full responsibility for results or statements, however, is taken by the author.

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Evaluation of Effective Radiant Heating Surface and Application of the Stefan-Boltzman Law to Heat Absorption in Boiler Furnaces

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In the first section of this paper a method has been developed for converting well-known types of furnace heat-absorbing surfaces into an equivalent effective radiant heating surface. It concerns the evaluation of this effective radiant heating surface as controlled by (1) various spacings of waterwall tubes, (2) various kinds of blocks on the waterwall tubes, (3) various amounts of slag on the waterwall tubes or blocks, and (4) the emissivity of the waterwall surface exposed to the flame. The second section deals with the fraction of heat supplied to the furnace that is given up by the gases in the furnace which is solved most simply and practically by the use of the Stefan-Boltzman radiation law.

Results computed in this manner have been compared with results reported in a paper entitled, "An Experimental Investigation of Heat Absorption in Boiler Furnaces."[2] It is found that for large pulverized coal, and oil-fired and gas-fired furnaces, the assumption of an effective furnace emissivity of unity leads to results which are in substantial agreement with experimental data. For stoker-fired furnaces in general, the effective furnace emissivity appears to be less than unity.

1—EFFECTIVE RADIANT HEATING SURFACE

OF THE number of methods offered for the evaluation of effective radiant heating surface in steam-boiler furnaces, the most logical method appears to be that involving in part the work of Hottel.[3,4] Briefly this method is based on the fact that radiant-heat transmission is governed by the laws of absorption and reflection. Thus, Lambert's cosine law is applicable, that is, a rough or corrugated "black" surface will not receive any more radiant heat than a similar plane or flat surface. Although this principle has been contested by some investigators it is generally accepted.

In this work we are interested in the areas in so far as they effect the radiant-heat transfer. Therefore, we may adopt as a standard unit of area, a square foot of effective radiant heating surface (water- or steam-cooled surface) which absorbs a maximum amount of radiant heat. If our premises are based on the Stefan-Boltzman law this unit square foot of radiant heating surface must necessarily be defined as a square foot of plane surface, perfectly clean, having an emissivity of unity, and so cooled by water or steam that its temperature is not greater than that of the water or steam which is cooling it and absorbing heat.

Radiation to this unit surface from a black body or unity emissive surface at various temperatures will be as shown in Fig. 1 which was derived from the Stefan-Boltzman law. This curve does not change appreciably for temperatures of the absorbing radiant heating surface varying from 0 to 1000 F abs. Hence, the standard unit square foot of effective radiant heating surface may be further defined as that square foot of surface which would absorb an amount of heat as indicated by Fig. 1 when receiving radiation from a black-body source at the given temperature.

Using this latter definition, any area having less radiant-heat absorption per square foot may be converted into an equivalent smaller area having the defined unit radiant-heat absorption per square foot by the use of multiplying terms which take into account the divergence of the conditions of the actual surface from those of the theoretically defined surface.

In practice, radiant-heat surfaces differ from the defined unit surface in that they are formed of metal tubes, often spaced at intervals, separated by refractory, and perhaps are covered by steel, cast-iron, or refractory-faced blocks. Various amounts of slag will form on the surfaces thus causing them to be less effective in absorbing heat. In addition the surfaces will have an emissivity less than unity.

If A is the projected overall waterwall area under these latter conditions and A_{rhs} is the effective radiant heating surface as defined, then

$$A_{rhs} = AF_AF_CF_SF_E \qquad [1]$$

where F_A is an area factor, F_C is an average conductivity factor, F_S is a slag factor, and F_E is an emissivity factor.

The area factor F_A is used to reduce overall surface to an equivalent effective radiant surface, assuming the surfaces clean and of perfect heat conductivity. The average conductivity factor F_C is used to take into account the fact that for refractory-

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faced blocks the heat absorbed by the waterwall is less than for blocks consisting of a material of perfect heat conductivity. That is, if a refractory face on a 1-sq ft block diminishes the heat transfer one-half, this surface is equivalent to 0.5 sq ft of defined effective radiant heating surface. It can be shown that this F_C-factor varies with furnace temperature. In this paper, however, a mean value for the average furnace temperature has been assumed. The value will be sufficiently correct for the present until more data becomes available. The slag factor F_S is similar to F_C, but is dependent for its value on the thickness of slag present. The emissivity factor F_E is used to take into account the fact that the emissivity of the surface is less than unity.

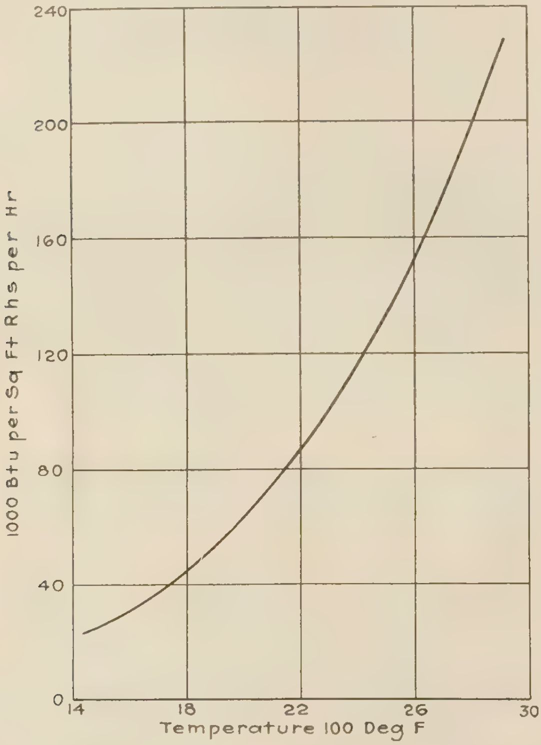

Fig. 1 Radiation Based on the Stefan-Boltzman Law. Emissivity = 1

Thus, effective radiant heating surfaces are computed on a basis of radiant-heat-absorbing ability, and all furnace waterwalls, no matter what their construction, may be reduced to a number of defined unit effective radiant heating square feet of surface by suitable selection of correction factors.

Area Factor F_A

The selection of these factors must be based upon existing data. Values of F_A may be mathematically calculated for all shapes and constructions of waterwalls. Fig. 2 shows curves taken from Hottel[3] for various bare-tube arrangements. Ordinarily curve 2, Fig. 2, is the only curve of those given that is used to determine F_A in boiler furnaces. The curves hold without significant error for all values of the dimension T for one row of tubes from zero to infinity.

The foregoing is applicable to completely bare tubes. Occasionally it is necessary to obtain F_A-values for bare tubes half recessed in refractory. It is necessary to solve for this separately using the method of Hottel as given in Fig. 4 of his first paper,[3] which will not be dealt with here. However, it is readily possible to draw up a curve similar to Fig. 2 for this special case.

Fin tubes may be handled in the same way when recessed in refractory and when the fins do not overlap or touch each other as is sometimes the case. Here the edges of the fins are to be considered as the heat-absorbing surface extremities. If these fin tubes are located at a distance from a refractory wall, Fig. 2 would be used. The heat-transfer coefficient of the metal of the fins is assumed to be sufficiently great so that the fins absorb as much heat per unit of exposed area as the tube wall does. This assumption is open to question but is adopted in view of the lack of data indicating otherwise. On this basis a fin tube has an effective diameter equal to the distance between the extreme edges of the fins. Where the tubes or fins touch each other, F_A must equal unity as L/D in Fig. 2 will equal unity.

For reasons similar to those stated for the fins of fin tubes, bolted-on or shrunk-on metal blocks with either plain or refractory faces have been assumed to have an effective area equal to the area of the face of the block. Where the blocks are adjacent to each other, F_A is equal to unity. The reduction in heat transfer caused by the joint between the block and the tube is taken into account in the F_C-factor.

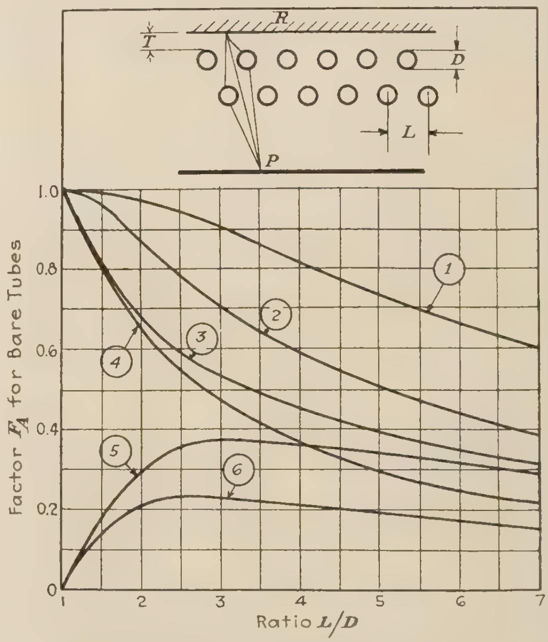

Fig. 2 Values of the F_A-Factor for Bare Tubes. These Values Hold for All Values of T Without Appreciable Error

(Curve 1—total to two rows when two are present. Curve 2—total to one row when only one is present. Curve 3—total to first row when two are present. Curve 4—direct to first row. Curve 5—total to second row when two are present. Curve 6—direct to second row. R denotes refractory. P denotes radiating plane. When two rows are present they are spaced so that the tubes form equilateral triangles. The tubes are considered to be perfect absorbers. The curves give the fraction (F_A) of total plane incident radiation absorbed either by the total tube, including reradiation, or by the tube directly, excluding reradiation by the refractory. For ordinary boiler-furnace calculations curve 2 only is used.)

The factor F_A is most simply defined as that factor by which it is necessary to multiply projected overall clean waterwall surface (including perfect conducting tubes and exposed refractory between the tubes) in order to obtain the effective exposed radiant surface. This effective surface is not appreciably different from the projected tube surface except where there are reradiating refractory walls in back of the waterwall tubes.

Conductivity Factor F_C

The conductivity factor F_C is introduced to take care of the heat resistivity of the wall. This factor is necessary because, although exposed areas of a bare metal block and a refractory-faced metal block may be equal, i.e., F_A is identical, these areas will absorb different amounts of heat by radiation from the furnace. Therefore, F_C is a ratio between the actual heat conducted and the maximum amount of heat that could be conducted, other conditions remaining constant.

E. G. Bailey[5] has given some figures indicating that the comparative heat-transfer rates of type-*BAG* refractory-faced blocks and type-*C* bare-faced blocks are 50,000 and 70,000 Btu per sq ft per hr. Thus F_C would be equal to 50,000/70,000 or 0.71. Other published material[6] and experimental data which cannot be included here would indicate that this value of the F_C-factor is somewhat high. After careful consideration of the meager data available the values of F_C in Table 1 were adopted.

TABLE 1 CONDUCTIVITY FACTORS

Type of surface	F_C
Bare tubes	1.00
Fin tubes	1.00
Bare-faced metal blocks on tubes	0.70
Refractory-faced metal blocks on tubes, type-*BAG*	0.35

Theoretically, F_C varies with furnace temperature but this correction is omitted as the present knowledge of the effect of refractory on waterwalls is so scanty as to render small corrections unimportant. There is urgent need of laboratory experiments to establish the F_C-values more correctly than the approximations given in this paper.

Slag Factor F_S

The slag factor F_S is introduced to take care of the reduced heat absorption of waterwalls due to accumulated slag. It is similar to the conductivity factor in its variation with furnace temperature, but in view of the meager data this correction seems superfluous. When the boiler furnace is first operated, F_S is equal to unity, it being assumed that the surfaces are uncoated with slag. During the course of time slag will build up on the waterwall surface until a certain average equilibrium thickness is reached. Therefore, it is possible to compute by the use of F_S the difference in heat-absorbing capacity of a boiler furnace when initially started and when operated under known conditions of wall cleanness.

The literature is almost devoid of data on this phase of the subject. Some information was obtained from two boiler tests[2] run under the direction of the A.S.M.E. Radiation Committee. In a stoker-fired boiler, in which the waterwall tubes comprise a separate screen boiler, records were available to show the amount by which the proportional evaporation of the screen boiler had been reduced from five days after initial operation up to 18 months after initial operation. While reliable records over such a period of time were not available for a pulverized-fuel-fired boiler having a similar separate screen boiler, a separate test was run on this boiler for which the slag was completely cleaned off of the side-wall tubes and the tubes wire-brushed to assure cleanness. Fig. 3 has been plotted from the results and is used to determine this slag factor. The derivation of these curves and the special case where bare tubes are located entirely in front of refractory are discussed in the Appendix to this paper.

[5] "Some Factors in Furnace Design for High Capacity," by E. G. Bailey, Trans. A.S.M.E., vol. 49–50, 1927–1928, paper FSP-50-78, p. 253.

[6] "Water-Cooled Furnaces for Underfeed Stokers," by O. De Lorenzi, *Combustion*, vol. 3 (new notation), May, 1932, p. 15.

Emissivity Factor F_E

There are very few data upon which the value of the emissivity factor F_E may be based for conditions as they exist within furnaces. This factor, as used by some investigators, was not

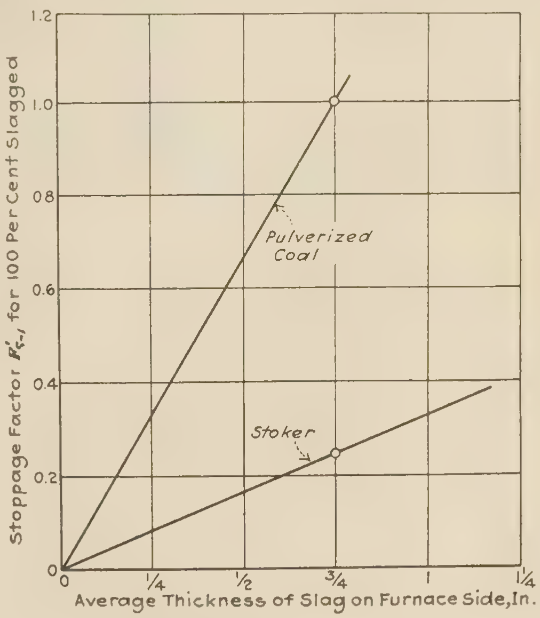

Fig. 3 Determinations of Slag Factor F_S

(For tubes half embedded in refractory, fin tubes, or waterwall block tubes, $F_S = (1 - xF'_{S-1})$ where x = fraction of the front of the tube that is slagged.)

clearly differentiated from the F_A-, F_C-, and F_S-factors, hence their data cannot be used. Wohlenberg has used a value of 0.95. There is every reason to believe that this factor is not far from unity.

Furnace Exit

Any radiant-heat ray from the furnace toward the exposed bottom boiler tubes at the furnace exit must strike tube surface, hence F_A will be unity. When these tubes are of the ordinary bare type, F_C is unity. The factor F_S may also be taken as unity when the tubes are relatively free of slag. Under these conditions the effective radiant heating surface A_{rhs} of the furnace exit will be equal to the product of the width and length of the tube bank.

In the special case of the double-ended boiler, it is obvious that the above will not hold. In this case it is suggested that the apex be considered nonexistent, that is, the area of the furnace exit be taken as the product of the length of the bottom side of the triangle of which the boiler tubes form two sides, and the width of the tube bank. The furnace volume used is then computed only up to this furnace exit.

Total Refractory Surface in the Furnace

Having obtained the effective radiant heating surface A_{rhs} by use of Equation [1], there will be an equivalent summation of areas A_{ref} defined by

$$A_{ref} = A - A_{rhs} \qquad [2]$$

The area A_{ref} is not refractory surface in the ordinary sense, no effective radiant heating surface; but rather one which may be considered to be the surface equivalent of a summation of non-absorbing particles of refractory located among the waterwall tubes.

Here a new concept is offered. For calculation of the total exposed refractory surface in the furnace in order to determine the fraction cold, it would appear reasonable to add this refractory area to the pure refractory occurring in the furnace as such. Thus, the total refractory area in a furnace will be the sum of the areas of any purely refractory walls plus the value ($A - A_{\text{rhs}}$).

The Stefan-Boltzman Law and Furnace Emissivity

It is known that radiation from an absolutely "black" body follows the Stefan-Boltzman law as shown in Fig. 1 which is comparatively simple and, for a hot surface completely surrounded by a cold surface, is expressed by

$$R = ECS[(T_1/100)^4 - (T_2/100)^4] \qquad [3]$$

where R = radiation, Btu per hr; C = black-body constant, 0.1723 Btu per sq ft per hr per (deg F abs)^4; S = area of hot surface, sq ft; E = effective emissivity, 1.00 for both surfaces black; T_1 = temperature of hot surface, F abs; T_2 = temperature of cold surface, F abs.

A furnace approximates these conditions only if the mass of hot gases or flames acts like a black body. To exactly what extent this is true is problematical. Small or microscopic black particles in the flame and the effect of reradiating refractory walls will cause the effective emissivity E to approach unity.

For purposes of simplification of the problem of furnace-heat-transfer calculation, T_1 of Equation [3] is taken equal to the furnace-exit gas temperature rather than a possibly higher mean radiating furnace temperature. The effective emissivity E based on the furnace-exit temperature will be higher than the effective emissivity E based on the higher mean radiating furnace temperature. Mathematically speaking, this is simply stating that the radiation of a body at a given temperature and given emissivity may be equal to that of the same body at a lower temperature and higher emissivity. Therefore, it is believed that the emissivity based on furnace-exit temperature may not be far from unity.

In regard to this particular matter the experimental data is not conclusive. Table 2 gives some results of calculation based on radiation pyrometer data taken in the course of the A.S.M.E. Radiation-Committee investigation.[2]

Table 3 contains some data given by Sherman.[7] Koessler[8]

TABLE 2 EMISSIVITY DATA

Station no.	Location	Avg flame emissivity[a]	Remarks
1	Grate	1.00	250,000 lb per hr rating
1	Furnace	0.75	250,000 lb per hr rating
2	Middle furnace	1.00	Same at all loads
2	Top of furnace	0.80	Same at all loads
7	Middle furnace	1.00	Same at all loads
8	Middle furnace	0.68	Increases with load
10	Middle furnace	0.72	Same at all loads

[a] Calculated.

TABLE 3 CALCULATED EMISSIVITY OF PULVERIZED ILLINOIS COAL BURNED IN SPECIAL EXPERIMENTAL FURNACE

Distance from burner, ft.	3.5	7.5	11.5
Emissivity, 10-ft flame depth	0.96	0.85	0.83
Emissivity, 20-ft flame depth	1.00	0.98	0.97

[7] "Burning Characteristics of Pulverized Coal and Radiation From Their Flames," by R. A. Sherman, *Combustion*, vol. 5 (new notation), December, 1933, p. 30.

[8] "Messungen der Flammenstrahlung in Dampfkesselfeuerungen," *Zeitschrift der Bayerischens Revisions-Vereins*, vol. 34, 1930; no. 6, March 31, pp. 75–78; no. 7, April 15, pp. 95–98; no. 10, May 31, pp. 146–149; no. 12, June 30, pp. 175–178; no. 13, July 15, pp. 184–187.

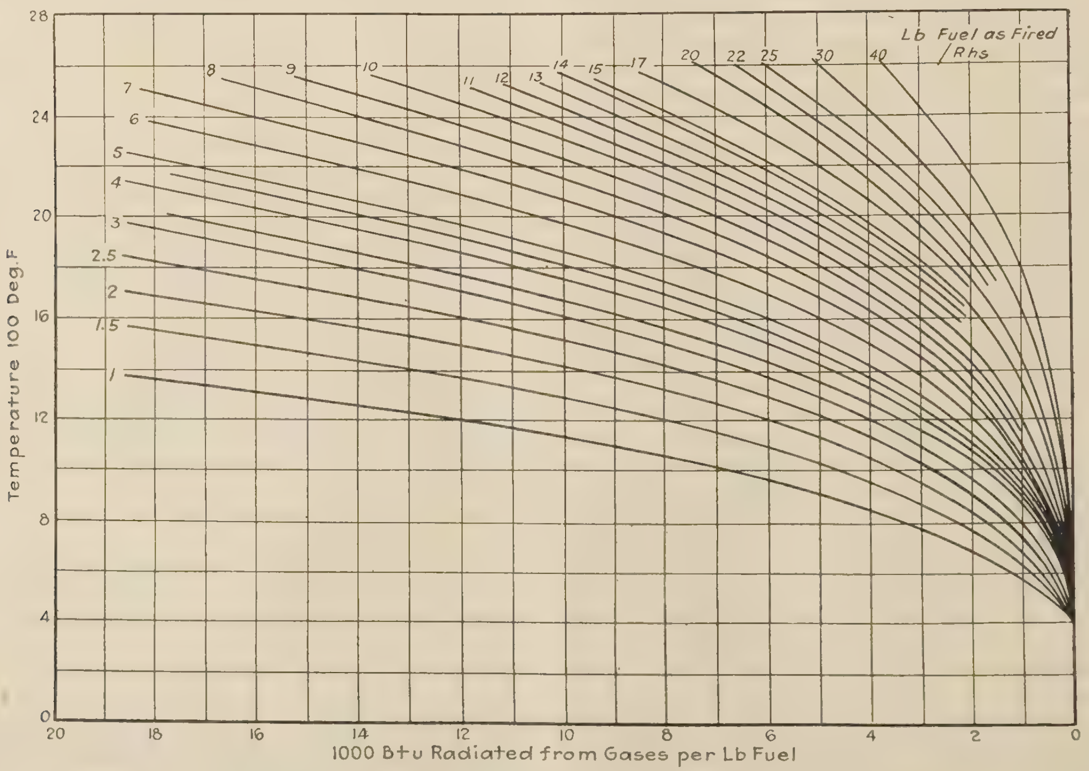

FIG. 4 HEAT RADIATED FROM GASES PER LB OF FUEL BURNED BASED ON THE STEFAN-BOLTZMAN LAW
(Lb fuel burned equals lb fuel as fired if consumption is assumed to be complete.)

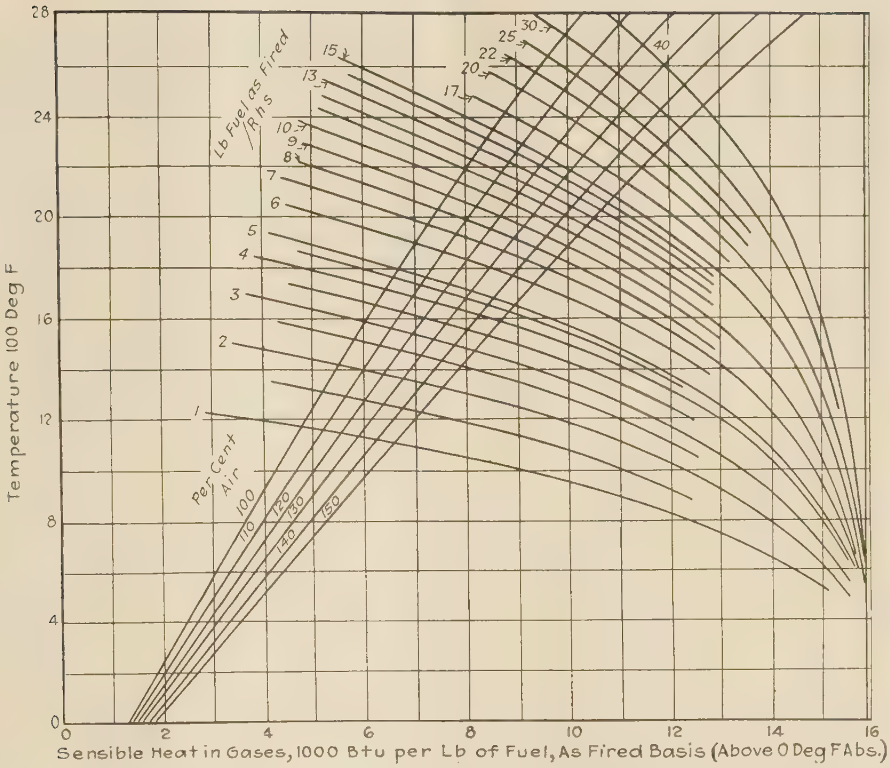

FIG. 6 CALCULATION OF μ_L BY MEANS OF THE METHOD ADVANCED IN THIS PAPER
(Example for test No. 1 on boiler No. 1 given in the A.S.M.E. Radiation-Committee report.[2])

has also given considerable data on several different types of furnace in which the values of emissivity very often reach unity. It is not possible to give his data here.

2—APPLICATION OF THE STEFAN-BOLTZMAN LAW TO FURNACE-HEAT-TRANSFER CALCULATIONS

Utilizing the preceding information, a method was developed for the solution of the problem of furnace radiation on the basis

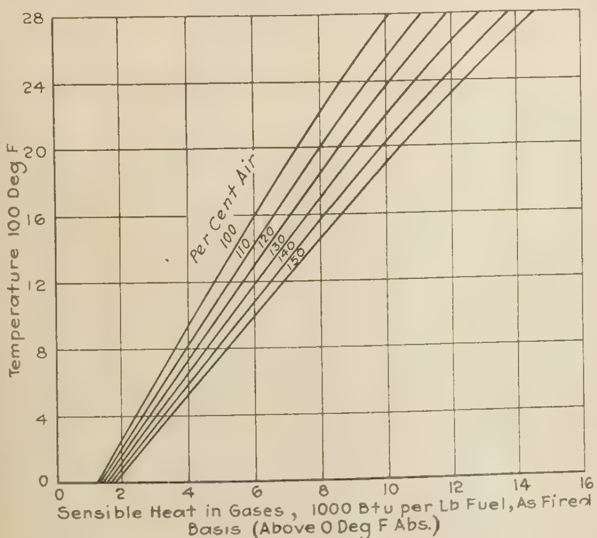

FIG. 5 HEAT-CAPACITY CURVES

that each square foot of effective radiant heating surface will absorb heat according to the Stefan-Boltzman law using an emissivity of unity, and that the furnace-exit temperature may be used as the mean radiating furnace temperature.

Fundamentally, the relation between the radiation to the furnace walls, furnace temperature, and heat in the gases leaving the furnace, is based upon a furnace heat balance which may be expressed by

$$Q_U + Q_A = Q_R + (Q_{GF} - Q_G) \qquad [4]$$

or

$$Q_U + Q_A + Q_G - Q_{GF} = Q_R \qquad [5]$$

where Q_A = heat of preheated air, Q_G = heat of gases at 80 F, Q_{GF} = heat of gases at furnace temperature, Q_R = heat radiated, Q_U = low heat value of fuel. To make use of Equation [5], the effective radiant heating surface in the furnace must be determined according to the principles given in Part 1 of this paper.

It will be observed that the principal differences between the proposed method and older schemes of applying the Stefan-Boltzman law to furnace-heat-transfer calculations lie in the manner of evaluating effective radiant heating surfaces and in the use of an emissivity of unity. A third difference lies in the practical method of graphical calculation now to be explained.

In Fig. 1 furnace temperature is plotted against heat received by radiation per square foot of effective radiant heating surface. It is desirable to introduce the combustion rate expressed as pounds of fuel burned per square foot of effective radiant heating surface. Now

$$\frac{\text{(Heat radiated per hr/rhs)}}{\text{(Lb fuel burned per hr/rhs)}} = \frac{\text{heat radiated}}{\text{lb of fuel burned}} \qquad [6]$$

By use of Fig. 1, Equation [6] and various values of pounds of fuel burned per square foot of effective radiant heating surface, the curves of Fig. 4 may be obtained. Since the effective radiant heating surfaces will absorb only a given amount of heat at any furnace temperature, this merely states that if the combustion rate is doubled only half as much heat can be absorbed per pound of fuel supplied. This will hold only if the furnace temperature remains the same. Actually the smaller quantity

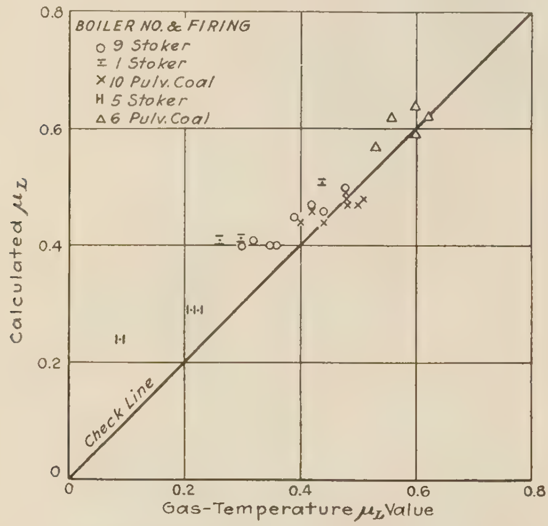

FIG. 7 COMPARISON BETWEEN RESULTS OBTAINED BY THE METHOD ADVANCED IN THIS PAPER AND THOSE OBTAINED FROM EXPERIMENTAL DATA

(Experimental data and boiler numbers are those given in the A.S.M.E. Radiation-Committee report.[2])

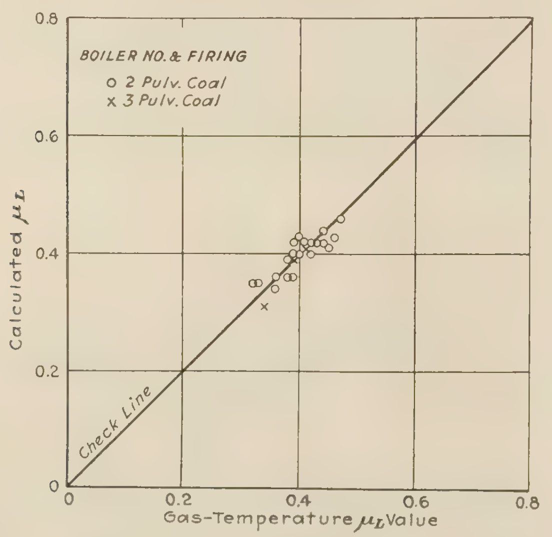

FIG. 8 COMPARISON BETWEEN RESULTS OBTAINED BY THE METHOD ADVANCED IN THIS PAPER AND THOSE OBTAINED FROM EXPERIMENTAL DATA

(Experimental data and boiler numbers are those given in the A.S.M.E. Radiation-Committee report.[2])

of heat absorbed by radiation causes the furnace temperature to rise and hence more heat will be radiated until a balance is reached.

To use Fig. 4 in calculations the procedure is as follows: Using a sensible-heat curve for the flue gases of the particular fuel, such as Fig. 5, the furnace temperature is found on both Figs. 4 and 5 such that the effective or low heat value per pound of wet fuel, plus the heat of preheat in the air, plus the heat in the gases of combustion at 80 F, all being the heat input per pound of wet fuel, minus the heat in the gases at the furnace temperature from Fig. 5, is equal to the heat radiated per pound of wet fuel in Fig. 4.

In order to eliminate the time involved in a trial-and-error solution, Fig. 4 should be drawn on tracing paper. It can then be superimposed on Fig. 5 as shown in Fig. 6, the origin on the heat-radiated curve being placed on the heat-capacity-curve base line at a value equal to the total heat input (using low heating value) to the furnace per pound of fuel as fired. At the intersection of the curve for per cent supplied air in the furnace and the curve for pounds of fuel per square foot of effective radiant heating surface, the furnace temperature, the heat in the gases at the furnace temperature, and the heat radiated may be obtained simultaneously. The value μ_L is then a ratio of the heat radiated to the heat input per pound of wet fuel, the

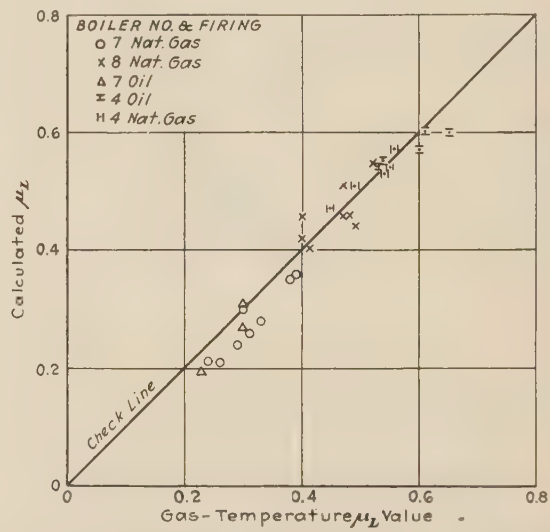

FIG. 9 COMPARISON BETWEEN RESULTS OBTAINED BY THE METHOD ADVANCED IN THIS PAPER AND THOSE OBTAINED FROM EXPERIMENTAL DATA

(Experimental data and boiler numbers are those given in the A.S.M.E. Radiation-Committee report.[2])

subscript $( )_L$ denoting that μ has been based on the low heat of the fuel.

Values of μ_L using this method have been computed for the tests given on ten boiler furnaces in the A.S.M.E. Radiation-Committee report.[2] These computed μ_L-values are compared with gas-temperature μ_L-values determined in that report[2] in Figs. 7, 8, and 9. The boiler numbers correspond to the boiler numbers of the report.[2]

It will be observed that results are satisfactory, for all types of fuels and firing, excepting the results for the stoker-fired furnaces which are high.

An attempt was made to derive empirical correction factors K_X that would modify the results obtained by this method in order to bring them more in line with the observed A.S.M.E. Radiation-Committee data.[2] The derived correction factors appear to be best embodied in the curves given in Figs. 10 and 11. These are to be applied by multiplying the obtained μ_L-values by the correction factor K_X. The modified μ_L-values thus obtained are compared in Figs. 12 and 13 with gas temperature μ_L-values given in the A.S.M.E. Radiation-Committee report.[2]

It will be seen that the modified results represent the data rather closely considering the simple correction used. The multiplying correction factor is unity for oil and gas firing except for all-refractory furnaces, in which case the mean furnace temperature is probably somewhat higher than the furnace-exit temperature due to the lack of waterwall surfaces available for absorbing heat in the lower portion of the flame. For this same reason the correction factor for pulverized coal approaches unity for furnaces in which the waterwall surface is not a large fraction of the total surface. The correction factor for stoker firing depends upon fraction cold in very much the same manner as has been shown by Wohlenberg in his method.

The new method given is not difficult. The greatest amount of time required in obtaining solutions by its use will be used in plotting a heat-capacity curve and in computing the effective

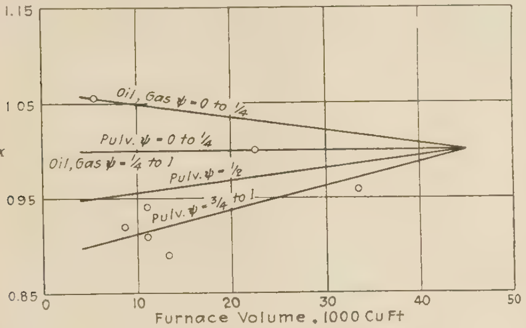

Fig. 10 Correction Factor K_X for μ_L — Suspension Firing
(Fraction cold ψ = effective radiant heating surface/total furnace surface less grate area.)

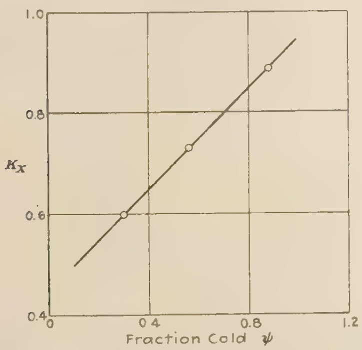

Fig. 11 Correction Factor K_X for μ_L — Stoker Firing
(Fraction cold ψ = effective radiant heating surface/total furnace surface less grate area.)

radiant heating surface. For extended use the former may be simplified by plotting heat-capacity curves for typical commercial fuels, which can also be used for boiler-design purposes. In routine work, a heat-capacity curve corresponding to the nearest fuel analysis may be used for calculations. There has been considerable agitation for a more rational method for evaluating effective radiant heating surface and it is possible that a method on the order of that which is given in this paper may be used in the future by boiler manufacturers to rate their product.

Comparison With the Broido and Orrok Methods

It can be shown that the Broido and Orrok methods are derivable from the preceding method of calculation. Using the Stefan-Boltzman law curve of Fig. 1 and a heat-capacity curve as in Fig. 5, other types of curves may be plotted. One type is shown in Fig. 14. This curve was obtained by assuming a furnace temperature from which the heat radiated per square foot of effective furnace surface was found in Fig. 1 and the value of μ_L for a particular per cent air and heat input to the furnace per pound of fuel was found from Fig. 5. Dividing the heat radiated by μ_L gives the heat liberated per square foot of radiant heating surface for the particular conditions. This was

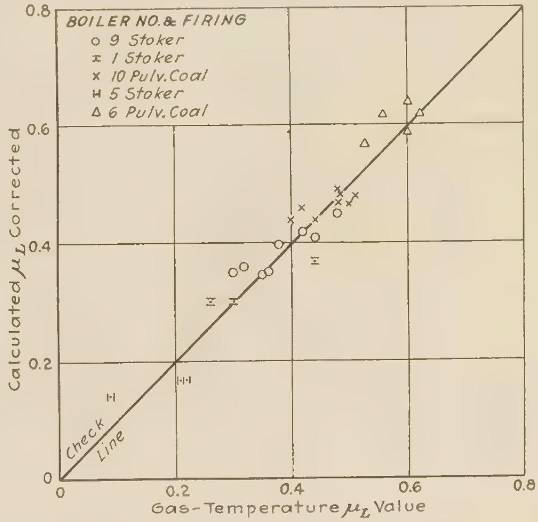

Fig. 12 Comparison Between Corrected Results Obtained by the Method Advanced in This Paper and Those Obtained From Experimental Data
(Experimental data and boiler numbers are those given in the A.S.M.E. Radiation-Committee report.[2])

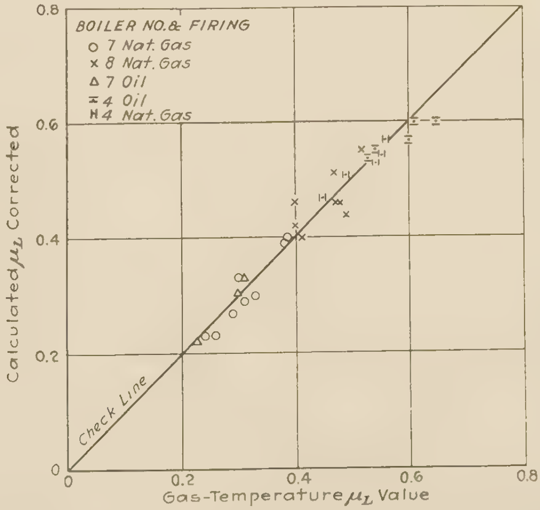

Fig. 13 Comparison Between Corrected Results Obtained by the Method Advanced in This Paper and Those Obtained From Experimental Data
(Experimental data and boiler numbers are those given in the A.S.M.E. Radiation-Committee report.[2])

done for a number of values of furnace temperature and the results were plotted as in Fig. 14. Thus, this curve might have been used to determine μ_L according to the new method derived. However, it was not used because it is not as flexible as the curves previously given which were plotted on a slightly different basis.

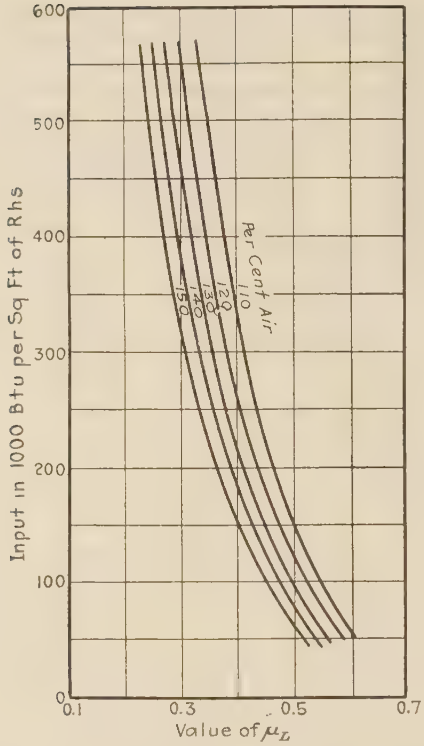

Fig. 14 Special Type of μ_L Chart

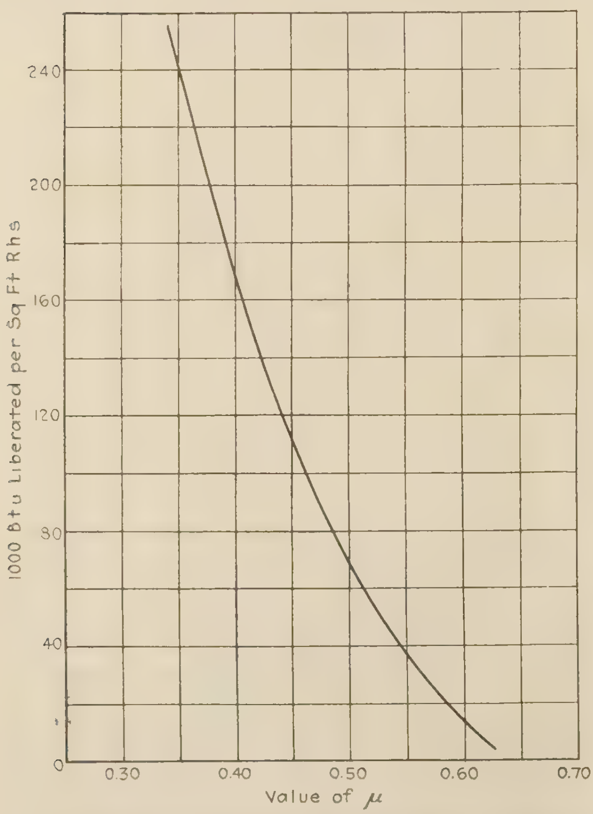

Fig. 15 Broido's Curve of Boiler-Furnace Radiation

An interesting fact is revealed by this type of curve. If it is compared with Fig. 15, which is a plot of the Broido curve, it will be seen that the Broido curve might be derived from the curves in Fig. 14 assuming a certain relation between the per cent of theoretical air for combustion and the rate of driving. A similar comparison for Fig. 16, which is a plot of the Orrok equation, shows that the Orrok equation might be similarly derived by assuming a given heat value input per pound of fuel.

Therefore, we might conclude that the Broido curve should be accurate for a furnace having a certain excess-air characteristic for various boiler loads and the Orrok equation is applicable to a fuel having a given heat value.

Accuracy of the Proposed Method

To the engineer who frequently has used empirical formulas for the calculation of furnace-heat transfer, the present attack based on the Stefan-Boltzman law may appear as a rational attack in that the empirical corrections, considering the diversity

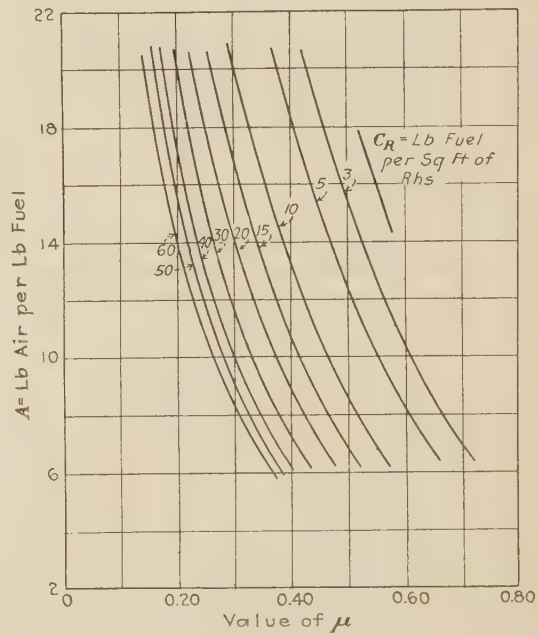

Fig. 16 Plot of Orrok's Equation for Boiler-Furnace Radiation

in types of fuels and firing arrangements, are not great. From the strictly theoretical viewpoint of the physicist it will appear to a large extent empirical. Whether the method be considered to be based upon a firm theoretical foundation or upon an empirical basis it is submitted as being in agreement with such experimental data as is available, as having sufficient simplicity for use in practical calculations, and as showing clearly from the graphical standpoint the influence of various operating factors on the result desired. It is probable that the method will give an average furnace-exit temperature within 75 F. It should be noted however that the data by means of which the method advanced in this paper has been checked were taken almost entirely from large furnaces. Therefore, application to small furnaces must be made with caution.

Study of the Effect of Combustion Conditions of Suspension Firing on Furnace-Outlet Temperature

A special study was made in order to determine the effect of fixed furnace characteristics such as fraction cold, furnace

volume, and furnace length, as well as such operating conditions as per cent air, temperature of preheated air, and energy-release rate. Using the method developed in this paper, the furnace-outlet temperature was computed for various arrangements

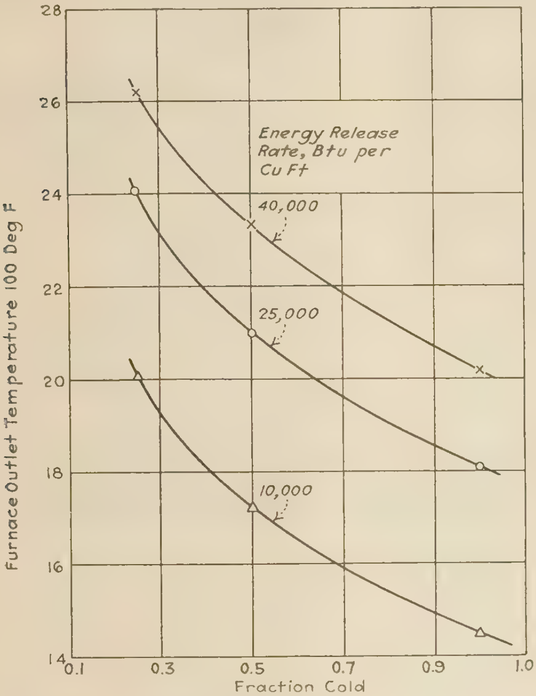

FIG. 17 EFFECT OF VARIATION OF FRACTION COLD ON FURNACE-OUTLET TEMPERATURE AT VARIOUS VALUES OF ENERGY-RELEASE RATE

(Air = 120 per cent, air temperature = 80 F, furnace volume = 8000 cu ft, and furnace length = 20 ft.)

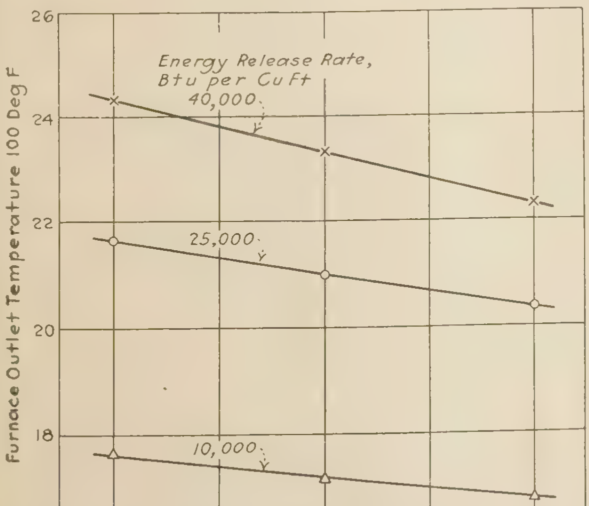

FIG. 18 EFFECT OF VARIATION OF PER CENT AIR ON FURNACE-OUTLET TEMPERATURE AT VARIOUS VALUES OF ENERGY-RELEASE RATE

(Fraction cold = 0.50, air temperature = 80 F, furnace volume = 8000 cu ft, and furnace length = 20 ft.)

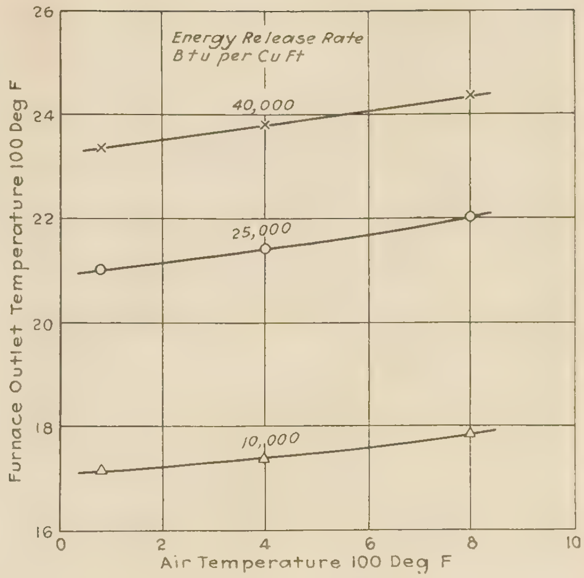

FIG. 19 EFFECT OF VARIATION OF AIR TEMPERATURE ON FURNACE-OUTLET TEMPERATURE AT VARIOUS VALUES OF ENERGY-RELEASE RATE

(Fraction cold = 0.50, air = 120 per cent, furnace volume = 8000 cu ft, and furnace length = 20 ft.)

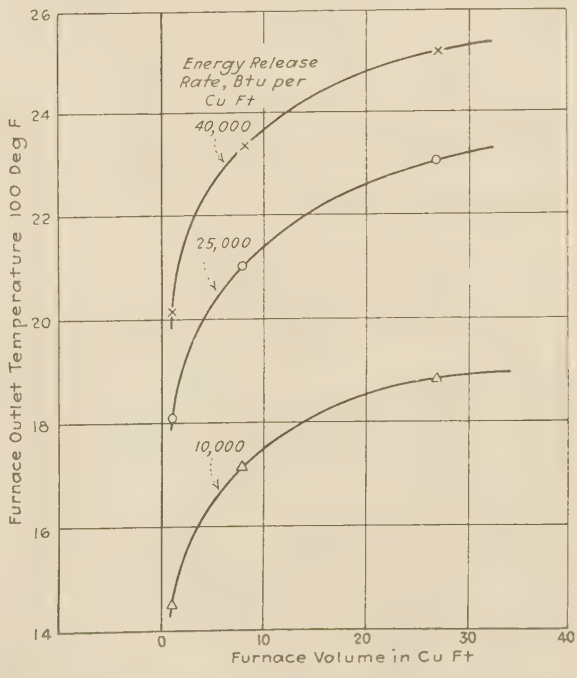

FIG. 20 EFFECT OF VARIATION OF FURNACE VOLUME ON FURNACE-OUTLET TEMPERATURE AT VARIOUS VALUES OF ENERGY-RELEASE RATE

(Fraction cold = 0.50, air = 120 per cent, air temperature = 80 F, furnace length = 10, 20, and 30 ft.)

and conditions of the six variables mentioned previously. Three values were chosen for each variable as shown in Table 4. Fraction cold is defined as the effective radiant heating surface in the furnace divided by the area of the entire furnace envelope.

The fuel chosen for these calculations was a representative

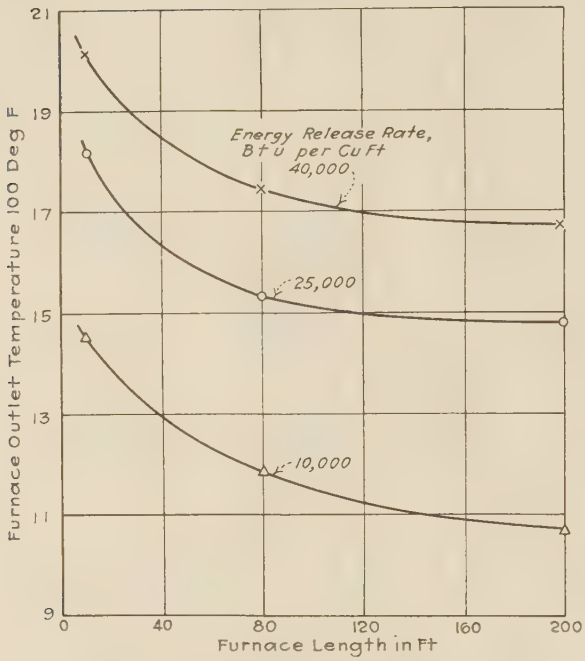

FIG. 21 EFFECT OF VARIATION OF FURNACE LENGTH ON FURNACE-OUTLET TEMPERATURE AT VARIOUS VALUES OF ENERGY-RELEASE RATE

(Fraction cold = 0.50, air = 120 per cent, air temperature = 80 F, furnace volume = 1000 cu ft.)

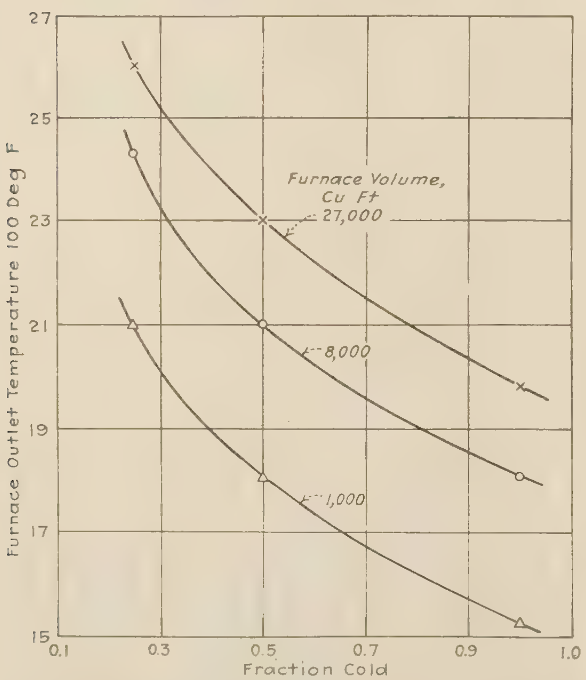

FIG. 22 EFFECT OF VARIATION OF FRACTION COLD ON FURNACE-OUTLET TEMPERATURE AT VARIOUS VALUES OF FURNACE VOLUME

(Air = 120 per cent, air temperature = 80 F, energy release rate = 25,000 Btu per cu ft, and furnace length = 10, 20, and 30 ft.)

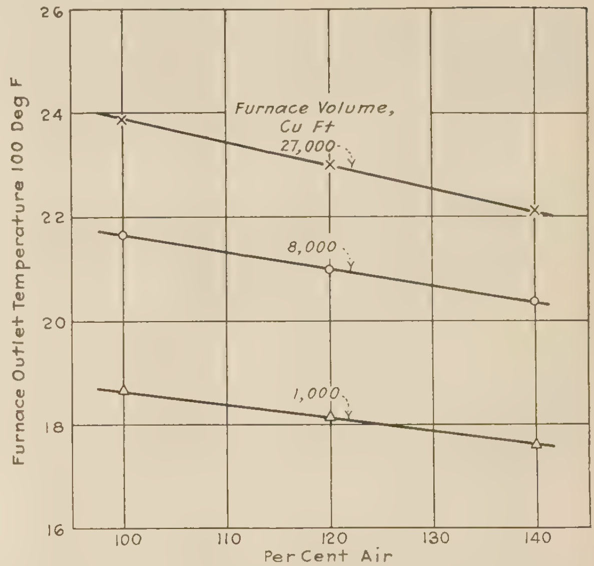

FIG. 23 EFFECT OF VARIATION OF PER CENT AIR ON FURNACE-OUTLET TEMPERATURE AT VARIOUS VALUES OF FURNACE VOLUME

(Fraction cold = 0.50, air temperature = 80 F, energy-release rate = 25,000 Btu per cu ft, and furnace length = 10, 20, and 30 ft.)

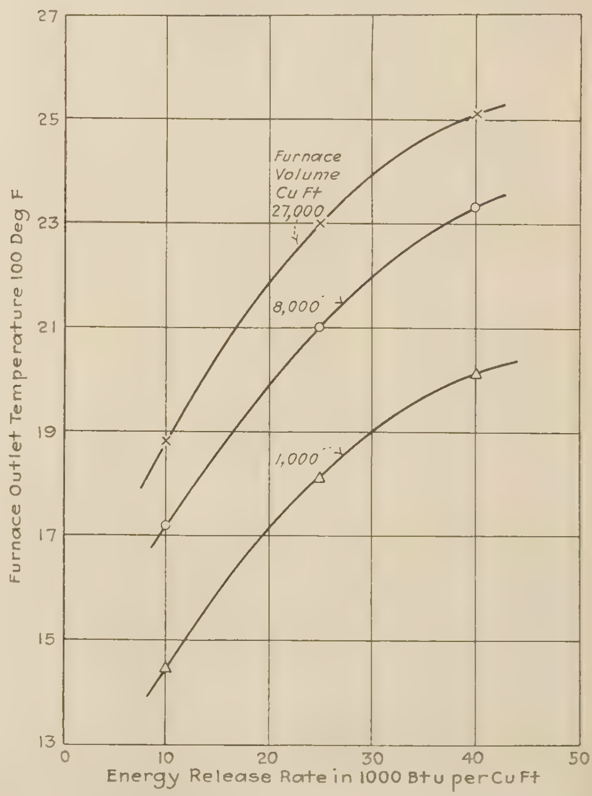

FIG. 24 EFFECT OF VARIATION OF ENERGY-RELEASE RATE ON FURNACE-OUTLET TEMPERATURE AT VARIOUS VALUES OF FURNACE VOLUME

(Fraction cold = 0.50, air = 120 per cent, air temperature = 80 F, furnace length = 10, 20, and 30 ft.)

Eastern bituminous coal, the heat-capacity curve of which is shown in Fig. 5.

All of the six groups of variables given in Table 4 were varied one at a time with respect to any other five held constant and furnace-outlet temperatures calculated by the method developed, except that the K_X correction factors were not applied. Therefore, the results, as far as the indication of absolute values is concerned, are slightly inaccurate, although the trends are correct.

Part of the results are presented in Figs. 17 to 26. From the shape of the curves, it is possible to draw certain general conclusions which are summarized in Table 5. These results agree with those which would be expected from qualitative theoretical considerations.

TABLE 4 VALUES CHOSEN FOR SIX VARIABLES GOVERNING FURNACE-OUTLET TEMPERATURES

Variable	Values chosen Low	Medium	High
1 Fraction cold	0.25	0.50	1.00
2 Per cent combustion air	100	120	140
3 Temperature combustion air, F	80	400	800
4 Energy-release rate, Btu per hr	10000	25000	40000
5 Furnace volume, cu ft	1000	8000	27000
6 Furnace length, ft	10, 20, or 30[a]	80	200

[a] Minimum furnace length is based on volume taken as a cube.

TABLE 5 QUALITATIVE EFFECT OF VARYING COMBUSTION CONDITIONS ON FURNACE-OUTLET TEMPERATURES

Increasing this factor	Effect on furnace-outlet temperature
1 Fraction cold	Decrease
2 Per cent combustion air	Decrease
3 Temperature of combustion air	Increase
4 Energy-release rate	Increase
5 Furnace volume	Increase
6 Furnace length	Decrease

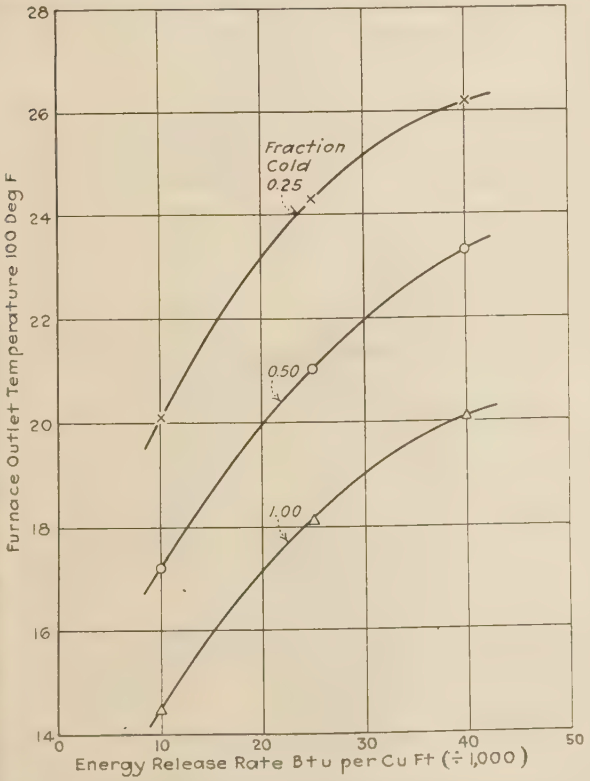

Fig. 25 Effect of Variation of Energy-Release Rate on Furnace-Outlet Temperature at Various Values of Fraction Cold

(Air = 120 per cent, air temperature = 80 F, furnace volume = 8000 cu ft, furnace length = 20 ft.)

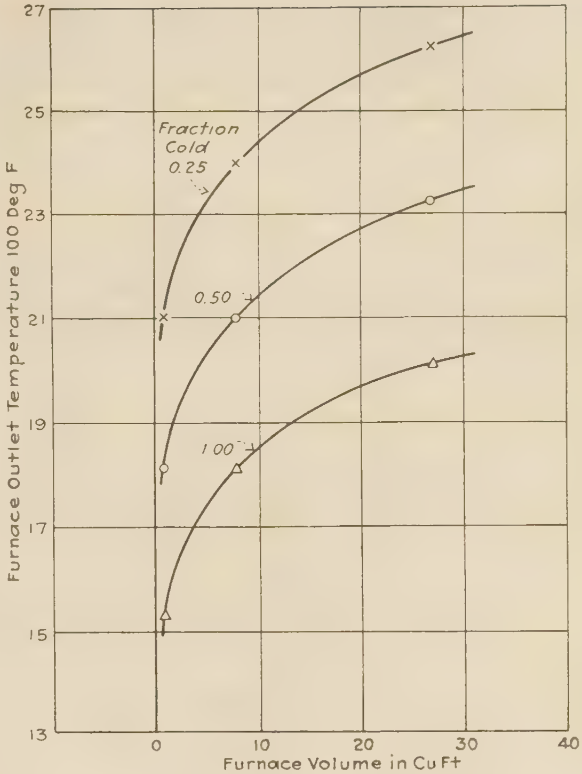

Fig. 26 Effect of Variation of Furnace Volume on Furnace-Outlet Temperature at Various Values of Fraction Cold

(Air = 120 per cent, air temperature = 80 F, energy-release rate = 25,000 Btu per cu ft, furnace length = 10, 20, and 30 ft.)

Appendix

Calculation of Slag Factors for Tubes by Use of Experimental Data

From the A.S.M.E. Radiation-Committee report,[2] the test of boiler No. 1, which is a stoker-fired unit, makes available data on a separate screen or waterwall boiler composed of side-wall tubes and a slag screen at the furnace exit. Data for the evaporation of the screen boiler, compared to the total evaporation, for clean tubes and for slag-covered tubes after 18 months' operation, are given in Table 6. The average per cent of slag on the furnace effective radiant-heating-surface area of 2027 sq ft of this boiler has been computed to be 55 per cent. In addition there are 394 sq ft of radiant heating surface at the furnace exit. It is assumed that the furnace-exit surfaces evaporate the same quantity of steam after long operation as

TABLE 6 EVAPORATION OF SCREEN BOILER NO. 1[a]

Total boiler evap, 1000 per hr	Screen-boiler evap, 1000 lb per hr Clean tubes	Slagged tubes	Evap ratio, slagged to clean tubes
120	67	58	0.865
140	75	67	0.895
180	93	84	0.903
200	102	92	0.900
240	120	108	0.900
260	129	114	0.885
300	147	124	0.845

[a] Data taken from a test of boiler No. 1 as given in the A.S.M.E. Radiation-Committee report.[2]

they did at initial operation, that is, they do not accumulate slag, the slag being accumulated only on the side waterwalls. From this an equation may be set up in which x = the slag factor and E = the original screen-boiler evaporation. Then, for initial operation

$$394 + 2027 = f(E) \qquad [7]$$

and for slagged operation

$$394 + 2027x = 0.9f(E) \qquad [8]$$

The value of 0.9 was taken as the approximate average reduction in evaporation of the screen boiler from Table 6. Solving the equation, $x = 0.88$. Thus, although the screen boiler evaporates

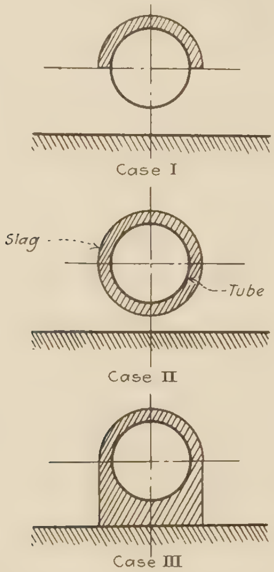

Fig. 27 Determination of F_S-Factor for Various Conditions

0.90 of its initial evaporation, because of the constant evaporation of the furnace-exit surface the screen waterwalls only evaporate 0.88 of their initial evaporation. If the wall is 55 per cent slagged with slag estimated 3/4 in. thick, it has a stoppage factor of 1.00 — 0.88 = 0.12. A wall 100 per cent slagged would have a factor of 0.12 (100/55) = 0.22. This value has been plotted as F'_{S-1} in Fig. 3. It is realized that the above solution is not correct inasmuch as it does not take into account convection at the furnace exit. More involved calculations would not appear to be justified by the data.

The second boiler from which tests are available in the A.S.M.E. Radiation-Committee report[2] is a pulverized-fuel-fired boiler with a bottom screen and side waterwalls on separate circulation. Here the effect of slagged waterwall tubes was obtained by comparing the evaporation of the screen or radiant boiler when slagged and when the side-wall tubes had been cleaned of slag and wire-brushed. Results are summarized in Table 7.

TABLE 7 EVAPORATION OF SCREEN BOILER NO. 2[a]

Total boiler evap, 1000 lb per hr	Screen boiler evap, 1000 lb per hr — Clean side of wall tubes	Slagged side of wall tubes	Evap ratio, slagged to clean tubes
150	72	64	0.890
200	91	82.5	0.906
250	107	98	0.906
280	114	103.5	0.908

[a] Data taken from a test of boiler No. 2 as given in the A.S.M.E. Radiation-Committee report.[2]

Boiler No. 2 is different from the stoker-fired boiler No. 1 in that the horizontal tubes of the screen boiler were not appreciably slagged and were therefore not cleaned. Their evaporation should remain constant.

It is estimated that in boiler No. 2 there were 340 sq ft of side-wall surface 30 per cent slagged with 3/4-in. thick slag. The horizontal bottom-screen tubes and the furnace exit together comprise 1083 sq ft of effective radiant heating surface. From Table 7, the heat transfer of the boiler was reduced 90.5 per cent because of the slag. For clean operation using the same method as for Equations [7] and [8]

$$1083 - 340 = f(E) \qquad [9]$$

and for slagged operation

$$1083 - 340x = 0.905f(E) \qquad [10]$$

from which $x = 0.60$.

Since this was for tubes 30 per cent slagged, tubes 100 per cent slagged would have a stoppage factor of (1.00–0.60) (100/30) = 1.33 which of course is impossible as 1.00 would be the highest value. Due to lack of data the factor has been taken as equal to 1.0 and is plotted as F'_{S-1} in Fig. 3.

Calculation of Slag Factors for Single Row of Tubes in Front of Refractory

The estimated per cent of slag is usually given as a percentage of the surface of the waterwall tubes on the furnace side. On this account a distinction must be made between tubes placed with the rear half imbedded in refractory and tubes placed en-

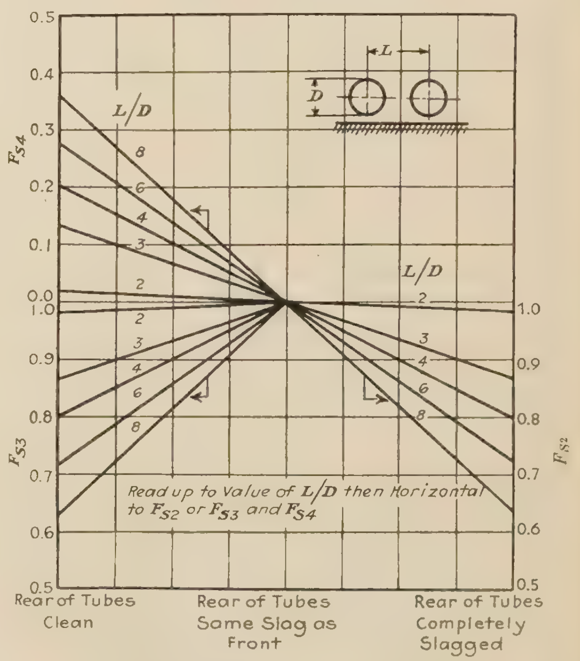

Fig. 28 Calculation of Slag Factors for Tubes in Front of Refractory

tirely in front of refractory. The first of these will be taken as the standard condition. The second case will necessarily be calculated somewhat differently, but will be related to the first case. The second case may be divided into three additional

divisions. The first is when the front of the tube is slagged and the rear of the tube is clean. The second is when the whole circumference of the tube is uniformly slagged. The third occurs when the front is slagged and the rear is completely slagged up. These cases are illustrated in Fig. 27.

TABLE 8 TABULATION OF RESULTS FROM HOTTEL'S CURVES

L/D	2	3	4	6	8
Heat direct to furnace side	0.66	0.47	0.37	0.25	0.08
Total heat to one tube	0.68	0.54	0.45	0.35	0.18
Diff. = heat to rear of tube	0.02	0.07	0.09	0.10	0.10
Fraction to front of tube	0.97	0.87	0.80	0.72	0.64
Fraction to rear of tube	0.03	0.13	0.20	0.28	0.36

It is necessary to set up a proportion between the amount of heat received by the front of the tube and the amount of heat received by the back of the tube. This has been done by the use of Hottel's curves of Fig. 2. The results are tabulated in Table 8.

Let F_S = the true slag factor, F'_{S-1} = the stoppage factor = (1.00 — slag factor), and x = the estimated fraction of front of tube slagged.

Consider the specific case where $L/D = 6$. Using Table 8 for the values of the heat to the front and the rear of the tubes and Fig. 3 for F'_{S-1} values, then for case 1

$$F_S = 0.28 - 0.72(1.00 - xF'_{S-1}) \qquad [11]$$

and for case 2

$$F_S = 1.00 - xF'_{S-1} \qquad [12]$$

and for case 3

$$F_S = 0.72(1.00 - xF'_{S-1}) \qquad [13]$$

This last case assumes that no heat is transferred to the back of the tube when it is completely slagged.

By letting $0.28 = F_{S4}$ and $0.72 = F_{S3}$ in case 1 and $0.72 = F_{S2}$ in case 3 a curve may be plotted to show the relation for various values of L/D as in Fig. 28. Equations [11], [12], and [13] then become

$$F_S = F_{S2}(1 - xF'_{S-1}) \text{ or } F_{S4} + F_{S3}(1 - xF'_{S-1}) \qquad [14]$$

which is the relation desired to define slag factors for tubes in front of refractory. The values for insertion in Equation [14] are obtained from Figs. 3 and 28.

Review of Methods of Computing Heat Absorption in Boiler Furnaces

By W. J. WOHLENBERG,[1] NEW HAVEN, CONN., and H. F. MULLIKIN,[2] NEW YORK, N. Y.

This paper contains results obtained by each of a number of methods of computing heat absorption in boiler furnaces. These are compared with the results reported by the A.S.M.E. Committee on Absorption of Radiant Heat in Boiler Furnaces.[3] Particular emphasis has been placed on the Broido, Orrok, and Wohlenberg methods. It is shown that the influence of various conditions as found by experiment are in substantial agreement with results expected from a theoretical consideration of the problem and those arrived at by certain well-known methods now in use.

Introduction

THE purpose of this paper is to endeavor to check, by such experimental data as may be available, the various formulas and methods for the calculation of boiler-furnace heat transfer as have been advanced by various investigators. It is not proposed to discuss either the derivation or the methods themselves to any extent as the original articles may be readily inspected.

These methods have never been checked quantitatively, with the exception of a few limited calculations on individual furnaces, principally because of the lack of experimental data which could be used for that purpose.

The report of the A.S.M.E. Committee on Radiation in Boiler Furnaces entitled "An Experimental Investigation of Heat Absorption in Boiler Furnaces,"[3] and hereafter referred to in this paper as the Radiation-Committee Report, furnished such data based on various sizes and types of furnaces fired in different manners. This report also has a bibliography which contains the titles and locations of the different papers describing the various methods of calculation to be utilized in this paper, as well as a short discussion of each.[4]

The form in which the original investigators have stated their results has in some cases required revision in order to enable comparisons to be made in a reasonable amount of time, such revision generally consisting of putting formulas in chart form. These revisions are described, in the first part of this paper, after which the calculations made on the basis of the experimental data are considered.

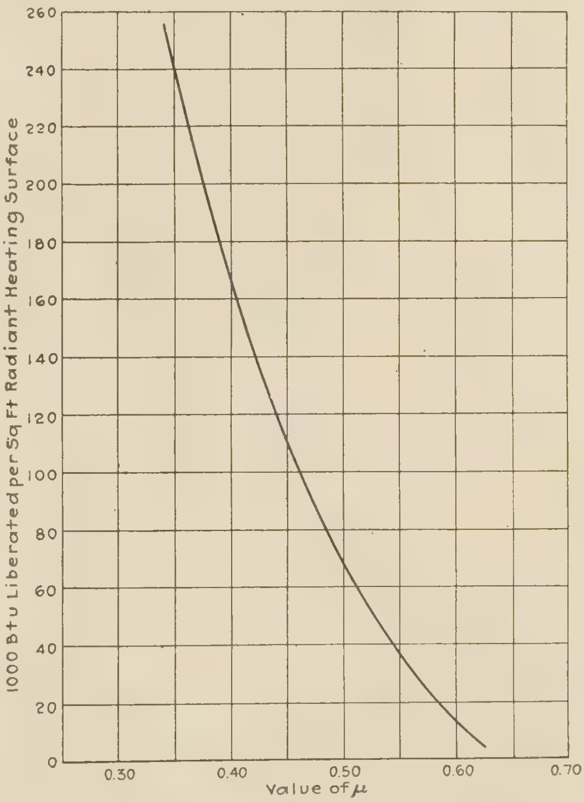

Fig. 1 Broido's Curve of Boiler-Furnace Radiation

Five methods of calculation have been investigated. These are (*a*) actual measurement (7, 12, 22);[4] (*b*) Broido method (7); (*c*) Orrok formula (20); (*d*) Wohlenberg method (1, 2, 3, 4, 5); and (*e*) Stefan-Boltzman law method. Method (*e*), for reasons stated later, is not included in the present paper.

Broido Method

Broido (7) has given a curve which represents the first important contribution to the subject. This curve, given in Fig. 1,

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[2] Research Fellow, Superheater Company-Yale University Fellowship. Jun. Member A.S.M.E. Dr. Mullikin is a graduate of Baltimore Polytechnic Institute and Johns Hopkins University, and the recipient of graduate degrees, including Ph.D. from Yale. He spent a year in the test department on boiler development at the General Electric Company.

[3] "An Experimental Investigation of Heat Absorption in Boiler Furnaces," by W. J. Wohlenberg, W. H. Armacost, C. W. Gordon, and H. F. Mullikin, Trans. A.S.M.E., vol. 57, paper RP-57-4.

Contributed by the Special Research Committee on Absorption of Radiant Heat in Boiler Furnaces for presentation at the Annual Meeting, December 2 to 6, 1935, of The American Society of Mechanical Engineers.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

[4] Numbers in parentheses correspond to similarly numbered references given in the bibliography of the A.S.M.E. Radiation-Committee report "Experimental Investigation of Heat Absorption in Boiler Furnaces."[3]

shows the relation between μ and heat input to the furnace per square foot of exposed projected radiant surface, where μ is defined as the ratio of heat given up from the furnace gases by radiation to the heat input to the furnace.

Orrok Formula

Orrok has given the following empirical formula derived from Hudson (39)

$$\mu = \frac{1}{1 + (A\sqrt{C_R}/27)} \qquad [1]$$

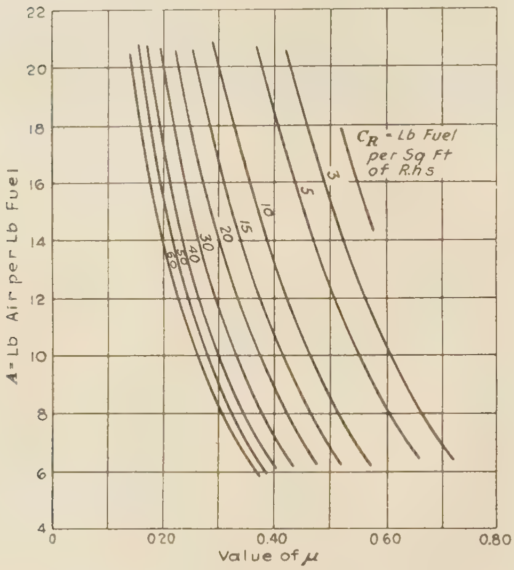

Fig. 2 Plot of Orrok's Equation for Boiler-Furnace Radiation

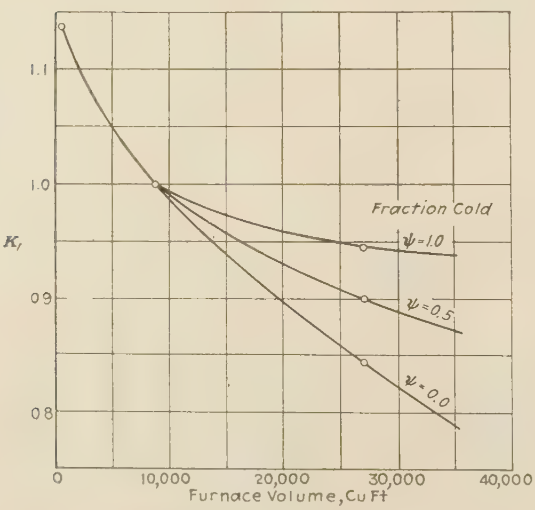

Fig. 3 Wohlenberg Curves—Influence of Furnace Volume

$$\left(\psi = \frac{\text{effective radiant heating surface}}{\text{total furnace surface less grate}}\right)$$

where μ = the fraction of liberated energy given up by gases in the furnace, A = lb of air per lb of fuel, and C_R = lb of fuel per sq ft of projected radiant heating surface. Fig. 2 shows Equation [1] plotted in the form of a series of curves for ready use. The Wilson, Lobo, and Hottel modification (45) is apparently not valid for the furnaces of steam boilers.

TABLE 1 CONDITIONS OF BASE SOLUTION FOR WOHLENBERG CURVES

1	Furnace volume	8,000 cu ft
2	Release rate	25,000 Btu per cu ft for pulverized-coal firing 40,000 Btu per cu ft for stoker firing
3	Excess air	120 per cent for pulverized-coal firing 140 per cent for stoker firing
4	Fraction cold[a]	unity
5	Coal	Illinois bituminous
6	Fineness	75 per cent through 200 mesh

[a] Fraction cold is defined as the effective exposed radiant heating surface in the furnace divided by the total exposed furnace surface (exclusive of the grate in stoker firing).

Wohlenberg Method

This is an extensive theoretical attack based upon the use of various physical constants rather than an empirical solution as

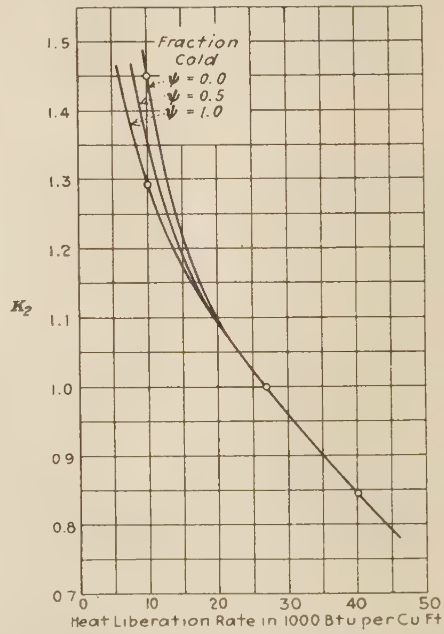

Fig. 4 Wohlenberg Curves—Influence of Liberation Rate. Liberation Rate Based on High Heat Value of Fuel + Air Heat

$$\left(\psi = \frac{\text{effective radiant heating surface}}{\text{total furnace surface less grate}}\right)$$

given in the two previous methods. The method is quite complex in that it attempts to evaluate, as far as possible, all the factors that enter into furnace design, in contrast to the simplicity of the empirical attacks which depend upon one or two variables. In applying this method use is made of a furnace heat balance and values of μ for various given conditions of furnace volume, energy-release rate, etc., which are given in several of the papers previously presented on this method.

It is possible, by use of this material to arrive at results by interpolation. For convenience in calculation, however, these data, together with some additional information, have been graphically restated in the present article in such a form as to enable a more rapid solution of the problem.

After trying several schemes of presentation it was decided to take as a base, the results for a furnace whose physical operating conditions were about in the middle of the range of physical characteristics selected for solution in the original papers. These conditions are given in Table 1.

Since the original publication of the Wohlenberg method,

better physical data have become available, especially in regard to the Stefan-Boltzman and gas-radiation coefficients. Thus, a Stefan-Boltzman coefficient of $\alpha = 0.1723$ Btu per sq ft per hr per (deg F abs)^4 (6), and the radiation values of Schmidt[5] are to be preferred.

The values of μ for the base furnace are henceforth designated as the F-factors. Under conditions of the base furnace defined in Table 1, the F-factors (μ-values) may be calculated using the

TABLE 2 VALUE OF F-FACTOR

Firing method	Method of calculation: With old coef	Method of calculation: With new coef	Coef ratio new/old
Pulverized	0.418	0.452	1.08
Stoker	0.258	0.311	1.20

old coefficients as given in the original papers (1, 2, 3, 4, 5) and also using the new coefficients mentioned in this paper. A comparison of results is shown in Table 2. It will be noted that the

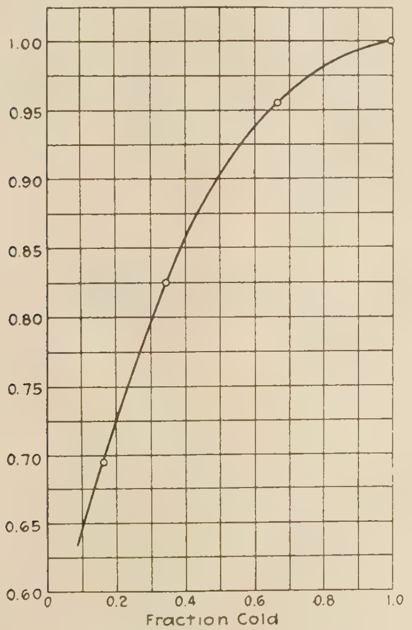

FIG. 5 WOHLENBERG CURVES—INFLUENCE OF FRACTION COLD

$$\left(\text{Fraction cold} = \frac{\text{effective radiant heating surface}}{\text{total furnace surface less grate}}\right)$$

values of the new coefficients are as much as 20 per cent greater than the old coefficients. The old values are valuable as a basis of comparison but the new F-factors should be used in computations.

Returning for the moment to the data tabulated by Wohlenberg and Lindseth (2) for pulverized coal, each of the factors of Table 1 was varied separately and the resultant ratios of the μ-values, termed K-factors, were plotted as shown in Figs. 3 to 7, inclusive. In order to obtain results within the desired degree of error, it was necessary to modify the curves somewhat as may be noted in the figures. The method of introducing this modification was determined by trial and error.

Using the curves obtained for the pulverized-coal furnace, a ratio was obtained, as shown in Figs. 8 and 9, between the results for pulverized-coal firing and for stoker firing.

The effect of preheated air was next noted in Fig. 10, where C is an additive increase in the value of μ.

[5] "Messung der Gessamtstrahlung des Wasserdampfes bei Temperaturen bis 1000 deg C," by E. Schmidt, *Foschung auf dem Gebiete des Ingenieurwesens*, vol. 3, 1932, p. 57.

Using some calculations made by the 1931 graduate power class at Yale University, the curve of Fig. 11 showing the effects of variation of particle size has been computed.

The curves give multiplying or additive correction factors to apply to the F-factor (μ-value) for the base or standard furnace. The procedure may be embodied in the equation

$$\mu = F\,K_1K_2K_3K_4K_5K_6K_7K_8 + C \qquad [2]$$

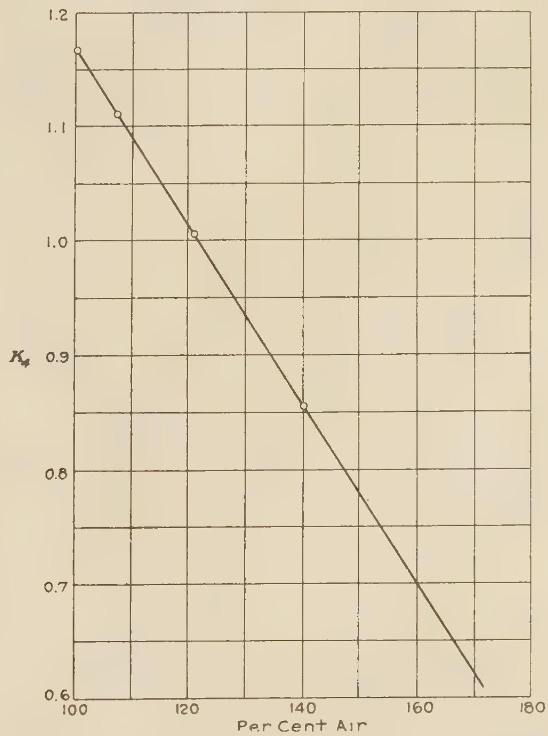

FIG. 6 WOHLENBERG CURVES—INFLUENCE OF EXCESS AIR

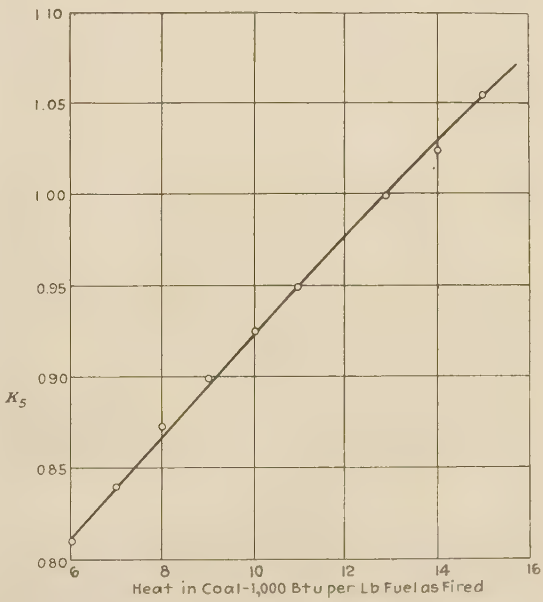

FIG. 7 WOHLENBERG CURVES—INFLUENCE OF FUEL HEAT VALUE

where F = 0.452 for pulverized-fuel firing and 0.311 for stoker firing. The K- and C-values are obtained from Figs. 3 to 11, inclusive, for the given conditions.

Thus, these figures give a series of correction curves for pulverized-coal and stoker furnaces which serve to determine μ for any condition of: (a) furnace volume, (b) heat-liberation rate, (c) amount of water-cooled surface in the furnace, (d) per cent air used for combustion, (e) heat value of fuel, (f) temperature of preheated air, and (g) coal fineness through 200 mesh.

One of the present authors has made a sample calculation (7), given in Table 3, for the value of μ for a gas-fired furnace as compared with pulverized-coal-fired and stoker-fired furnaces.

It will be seen that, according to the Wohlenberg method, μ

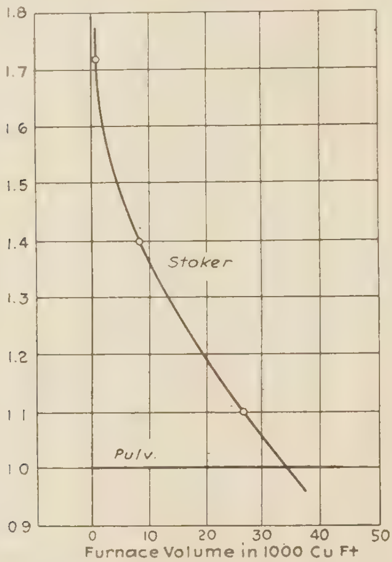

FIG. 8 WOHLENBERG CURVES—ADDITIONAL INFLUENCE OF FURNACE VOLUME

is much less for gas firing than for either stoker or pulverized-coal firing. As will be later shown, the experimental data at hand would seem to indicate otherwise. This results in the conclusion that the Wohlenberg method in its present form and as applied does not hold for gas or oil firing.

Equation [2] thus developed may be solved for any pulverized-coal or stoker furnace within 2 or 3 minutes, the only work involved being the selection of the appropriate K- and C-factors from Figs. 3 to 11, inclusive, and a simple slide-rule multiplication. From the value of μ obtained, the temperature of the flue gases leaving the furnace may be found by means of heat-capacity curves. These heat-capacity curves, Fig. 12, show the relation between the sensible heat of a given quantity of gas and its temperature and may be found by conventional combustion calculations using the ultimate analysis of the fuel.

The curves just developed will check values obtained by using the somewhat lengthy Wohlenberg equations within a maximum error of about ±0.03 in the value of μ. An advantage of the curves given in this manner is that they show clearly the effect of variation of any one of the factors.

HEAT RELEASE

Before discussing additional possible methods of calculation, it will be desirable to examine the experimental data given by the A.S.M.E. Radiation-Committee report[3] in connection with the Broido, Orrok, and Wohlenberg methods. In making such an examination, it is necessary to consider two quantities which

TABLE 3 COMPARISON OF μ FOR DIFFERENT FIRING METHODS AS COMPUTED BY THE WOHLENBERG METHOD

Firing method	Air, per cent	Furnace temp, F	μ
Pulv. coal	120	2230	0.40
Stoker	120	2386	0.36
Gas	120	2720	0.25

enter into the combustion process, namely, heat released in the furnace and the radiant heating surface in the furnace. There has been considerable diversity in the method of calculating these items in the literature.

In regard to the heat liberated in the furnace, some of the attacks on the problem have used an energy-release rate based on the calorific or high heat value of the fuel plus any sensible heat in the combustion air due to preheating.

From a theoretical point of view this is not thoroughly logical in that the latent heat required to vaporize the moisture due to hydrogen and water in the fuel is invaluable for raising the temperature of the gases or for radiation. In other words, instead of the high heat value, the low heat value should be used as in European practice, even though it may be somewhat indeterminate.

Thus, as far as furnace radiation is concerned, the low or net heat value should be used in calculations to determine the heat released per cubic foot of furnace volume, the heat released per pound of fuel supplied, and the value of μ.

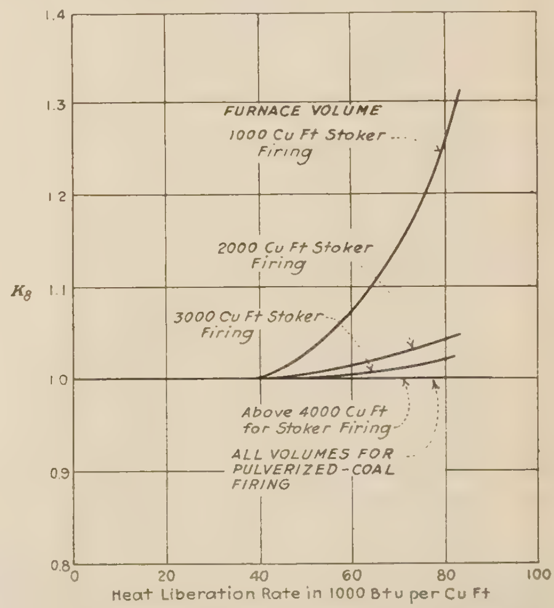

FIG. 9 WOHLENBERG CURVES—ADDITIONAL INFLUENCE OF LIBERATION RATE. LIBERATION RATE BASED ON HIGH HEAT VALUE OF FUEL + AIR HEAT

Various investigators have used the high or low heat value of the fuel rather indiscriminately. The Wohlenberg method made use of the high value and the Orrok method has made use of the low value although Orrok has compared his μ-values directly with values given in the Wohlenberg papers, thereby introducing confusion. Broido's procedure is not clear. The A.S.M.E. Radiation-Committee Report[3] gives experimental values of μ computed on both the high- and low-heat-value basis.

The heat supplied per pound of fuel fired should take into account incomplete combustion in any form.

In order to relate the various furnace-heat-transfer methods to each other, they have been reduced to a determination of μ, from which the furnace-exit temperature may be found through the use of heat-capacity curves. This μ in the case of the Wohlenberg

method has been based on the high heat value of the fuel, denoted as μ_H, hence the comparison with the experimental data will be made on this basis. The Orrok and Broido methods will be compared by a μ based on the low heat value of the fuel, denoted as μ_L. The μ-values calculated in either manner do not differ greatly, the maximum differences occurring, of course, for natural-gas firing.

Since the degree of completion of combustion in the various

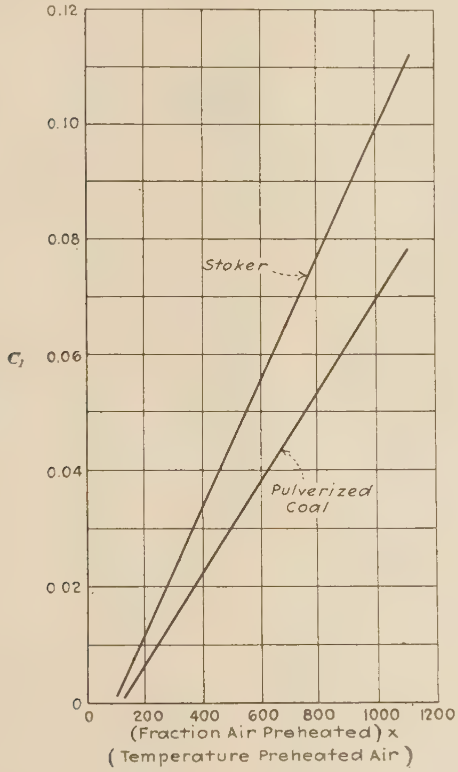

Fig. 10 Wohlenberg Curves—Influence of Preheat

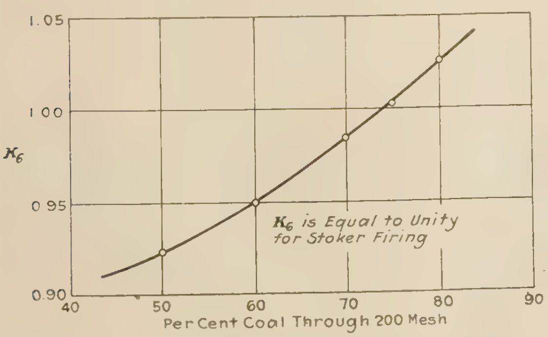

Fig. 11 Wohlenberg Curves—Influence of Pulverization Fineness

furnaces listed in the A.S.M.E. Radiation-Committee report[3] is not accurately known, it has seemed advisable to compare μ-values, computed by the various methods, on the basis of complete combustion. The gas-temperature μ-values of the report[3] were likewise computed on this basis and therefore the error in making this comparison should not be large.

Effective Furnace Radiant Heating Surface

The radiant heating surfaces in the furnace are those portions of the waterwalls and lower surface of the tubes of the boiler proper which "see" the gases in the furnace. Considerable difference of opinion exists on the exact method of evaluating this surface, due to different arrangements of bare tubes in front of refractory, bare tubes recessed in refractory, fin tubes, cast-iron or steel block covered tubes, and refractory-faced block-covered tubes.

For bare tubes the projected surface, that is, the diameter

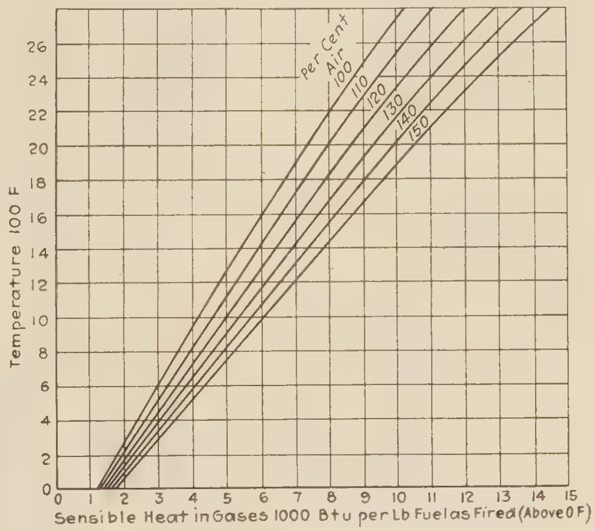

Fig. 12 Heat-Capacity Curves for Bituminous Coal Using Goodenough Specific Heats

times the length is often taken. Many use the surface based on half the circumference, the surface based on the whole circumference, and other possible combinations. In comparing or using any method, the manner of measuring the surface must be noted.

Obviously, the effective radiant heating surface should have a value independent of any method that may be used for the calculation of heat absorption. It is usually thought of as the equivalent plane surface which absorbs the same amount of heat as the cold surface or waterwall under consideration.

Most of the methods have given the procedure for the calculation of heat absorption, that is, the μ-value, without stating how to reduce the furnace walls actually found in practice to this plane-surface base.

In order to establish the values of the effective furnace radiant heating surfaces of the different boilers tested, usually denoted by rhs, it was necessary to decide upon a procedure which would be applicable to all types of furnace surfaces such as bare block, refractory-faced block, bare tube, and fin tube. This meant that certain assumptions had to be made, based upon such scanty information as could be discovered concerning the relative heat-absorbing capacities of such surfaces as refractory-faced blocks and fin tubes.

It is appreciated that a procedure so selected may not be acceptable to the proposers of the various formulas and methods, on the ground that they either recommend other procedures or had other procedures in mind when presenting their method.

However, the impracticability of calculating different effective radiant heating surfaces for each of the various furnaces is obvious and the results given in this paper must be accepted with this in mind.

For the purpose of this paper, the radiant heating surface has

TABLE 4 DETERMINATION OF FURNACE EFFECTIVE RADIANT HEATING SURFACES[a]

1 Boiler no.[b,c]	1	2	3	4	5	6	7	8	9	10	11	12	13
2 Total refractory surface	100	2985	3439	900	1136	142	1503	1224	180	184	2071	10.8	1940
Refractory-faced waterwall block:													
3 Overall surface	616	..	..	..	162	417	..	..	..	..	..	{0 / 42}	..
4 F_A	1.00	..	..	..	1.00	1.00	..	..	..	..	..	1.00	..
5 F_C	0.35	..	..	..	0.35	0.35	..	..	..	..	..	0.35	..
6 F_S	0.83	..	..	..	0.89	0.84	..	..	..	..	..	0.98	..
7 Effective surface	180	..	..	..	52	122	..	..	..	..	..	{0 / 14}	..
8 Additional refractory surface	436	..	..	..	110	295	..	..	..	..	..	{0 / 28}	..
Bare-faced waterwall block:													
9 Overall surface, front and rear	278	..	..	..	..	..	..	..	166	706	..	...	..
10 F_A	1.00	..	..	..	..	..	..	..	1.00	1.00	..	...	..
11 F_C	0.70	..	..	..	..	..	..	..	0.70	0.70	..	...	..
12 F_S	0.82	..	..	..	..	..	..	..	0.96	0.92	..	...	..
13 Effective surface	160	..	..	..	..	..	..	..	112	454	..	...	..
14 Additional refractory surface	118	..	..	..	..	..	..	..	54	252	..	...	..
15 Overall surfaces, sides	488	..	..	..	234	..	..	..	78	..	..	{125.5 / 83.5}	..
16 F_A	1.00	..	..	..	1.00	..	..	..	1.00	..	..	1.00	..
17 F_C	0.70	..	..	..	0.70	..	..	..	0.70	..	..	1.00	..
18 F_S	0.95	..	..	..	0.90	..	..	..	0.96	..	..	1.00	..
19 Effective surface	324	..	..	..	148	..	..	..	52	..	..	{125.5 / 83.5}	..
20 Additional refractory surface	164	..	..	..	86	..	..	..	26	..	..	0	..
Bare-tube waterwall:													
21 Overall surface, front	..	..	..	..	..	407	..	..	417	..	..	...	..
22 F_A	..	..	..	..	..	1.00	..	..	0.97	..	..	...	..
23 F_C	..	..	..	..	..	1.00	..	..	1.00	..	..	...	..
24 F_S	..	..	..	..	..	0.84	..	..	0.96	..	..	...	..
25 Effective surface	..	..	..	..	..	343	..	..	388	..	..	...	..
26 Additional refractory surface	..	..	..	..	..	64	..	..	29	..	..	...	..
27 Overall surfaces, sides	580	454	..	360	..	898	..	..	574	..	..	...	660
28 F_A	0.67	0.75	..	0.66	..	1.00	..	..	1.00	..	..	...	0.62
29 F_C	1.00	1.00	..	1.00	..	1.00	..	..	1.00	..	..	...	1.00
30 F_S	0.88	0.63	..	1.00	..	0.84	..	..	0.96	..	..	...	0.96
31 Effective surface	342	214	..	238	..	754	..	..	552	..	..	...	392
32 Additional refractory surface	238	240	..	122	..	144	..	..	22	..	..	...	268
33 Overall surface, rear	384	..	..	364	..	294	..	..	333	..	228	...	..
34 F_A	0.67	..	..	0.93	..	1.00	..	..	1.00	..	0.76	...	..
35 F_C	1.00	..	..	1.00	..	1.00	..	..	1.00	..	1.00	...	..
36 F_S	0.89	..	..	1.00	..	0.84	..	..	0.96	..	0.96	...	..
37 Effective surface	229	..	..	338	..	247	..	..	320	..	167	...	..
38 Additional refractory surface	155	..	..	26	..	47	..	..	13	..	61	...	..
39 Overall surface, bottom	..	860	860	..	..	57	..	..	..	945	314	...	700
40 F_A	..	0.75	0.75	..	..	1.00	..	..	..	0.77	0.76	...	0.65
41 F_C	..	1.00	1.00	..	..	1.00	..	..	..	1.00	1.00	...	1.00
42 F_S	..	1.00	1.00	..	..	1.00	..	..	..	0.92	0.96	...	1.00
43 Effective surface	..	645	645	..	..	57	..	..	..	670	229	...	455
44 Additional refractory surface	..	215	215	..	..	0	..	..	..	275	85	...	245
Fin-tube waterwall:													
45 Overall surface, front and rear	..	..	..	355	..	..	..	792	..	1618	..	...	..
46 F_A	..	..	..	1.00	..	..	..	1.00	..	0.63	..	...	..
47 F_C	..	..	..	1.00	..	..	..	1.00	..	1.00	..	...	..
48 F_S	..	..	..	1.00	..	..	..	1.00	..	0.92	..	...	..
49 Effective surface	..	..	..	355	..	..	..	792	..	938	..	...	..
50 Additional refractory surface	..	..	..	0	..	..	..	0	..	680	..	...	..
51 Overall surface, sides, top	..	..	..	800	..	..	..	800	..	2894	..	...	1060
52 F_A	..	..	..	1.00	..	..	..	1.00	..	1.00	..	...	1.00
53 F_C	..	..	..	1.00	..	..	..	1.00	..	1.00	..	...	1.00
54 F_S	..	..	..	1.00	..	..	..	1.00	..	0.92	..	...	0.96
55 Effective surface	..	..	..	800	..	..	..	800	..	2660	..	...	1016
56 Additional refractory surface	..	..	..	0	..	..	..	0	..	234	..	...	44
Furnace exit:													
57 Overall effective surface	394	438	438	364	408	360	357	475	555	432	360	...	330
Summation:													
58 Total refractory	1211	3440	3654	1048	1332	692	1503	1224	324	1625	2217	{10.8 / 38.8}	2497
59 Total effective waterwall surface	1629	1297	1083	2095	608	1883	357	2067	1979	5154	756	{125.5 / 97.5}	2193
60 Total furnace surface, excluding grate	2840	4737	4737	3143	1940	2575	1860	3291	2303	6779	2973	136.3	4630
61 Fraction cold (item 59/item 60)	0.57	0.27	0.23	0.67	0.31	0.73	0.19	0.63	0.86	0.76	0.25	{0.92 / 0.71}	0.47
62 Effective surface, screen boiler	1629	859	645	..	..	..	..	..	..	..	396	{125.5 / 97.5}	1471

[a] All surfaces are given in square feet.
[b] Boiler numbers 1 to 10, inclusive, are the same as listed in the Report of the Committee on Absorption of Radiant Heat in Boiler Furnaces, "An Experimental Investigation of Heat Absorption in Boiler Furnaces."[3]
[c] Data for boilers 11, 12, and 13 are taken from the literature.
NOTE: F_E taken equal to 1.00.

been computed on the basis given in an article by one of the present authors.[6]

This procedure makes use of the formula

$$A_{rhs} = A\, F_A F_C F_S F_E \qquad [3]$$

where A_{rhs} = equivalent effective plane radiant heating surface; A = the total overall plane area of the wall in question, regardless of the tube spacing, whether bare or block covered, etc.; F_A = an area factor to reduce overall surface to equivalent effective radiant heating surface, assuming all surfaces clean and of perfect conductivity; F_C = a conductivity factor to take into account the heat conductivity of various types of block-covered walls; F_S = a slag factor to account for reduction in heat transfer caused by slag coatings; and F_E = an emissivity factor.

The value of the factors for use in Equation [3] under various conditions are given by H. F. Mullikin[6] and not repeated here.

[6] "Evaluation of Effective Radiant Heating Surface and Application of the Stefan-Boltzman Law to Heat Absorption in Boiler Furnaces," by H. F. Mullikin, Trans. A.S.M.E., vol. 57, 1935, RP-57-2.

TABLE 5 DETERMINATION OF μ-VALUES BY DIFFERENT METHODS

2	39	40	41	42	43	44	45	46	47
Test	Heat absorbed by screen boiler, 10^6 Btu per hr	Heat to furnace surfaces based on screen boiler, 10^6 Btu per hr	μL based on screen boiler evap	Heat (high) input per unit rhs[a] 1000 Btu per hr	μ calculated by Broido method	Air-fuel rate R, lb air per lb fuel	Fuel-rhs rate C_R, lb wet fuel per unit rhs	μ calculated by Orrok method	μ calculated by Wohlenberg method
1–1	105	105	0.26	252	0.34	12.1	17.3	0.35	0.31
1–2	84	84	0.30	179	0.39	14.5	12.2	0.35	0.29
1–3	49	49	0.35	89	0.47	13.6	6.1	0.45	0.39
2–1	52.1	78.8	0.41	154	0.41	13.3	11.2	0.38	0.36
2–2	56.6	85.6	0.44	154	0.41	13.6	11.2	0.37	0.36
2–3	58.7	88.7	0.46	150	0.41	12.1	10.9	0.40	0.40
2–4	54.5	82.3	0.43	155	0.41	14.6	11.3	0.36	0.33
2–5	61.7	93.2	0.48	154	0.41	13.6	11.2	0.38	0.36
2–6	57.8	87.4	0.46	153	0.41	12.8	11.2	0.39	0.38
2–7	56.0	84.7	0.44	154	0.41	13.8	11.2	0.37	0.35
2–8	54.2	82.0	0.43	154	0.41	13.8	11.2	0.38	0.35
2–9	56.0	84.6	0.44	153	0.41	12.9	11.2	0.38	0.38
2–10	59.0	89.1	0.30	237	0.35	12.6	17.3	0.34	0.35
2–11	64.3	97.1	0.34	230	0.36	11.5	16.8	0.36	0.38
2–12	67.7	102	0.35	234	0.36	11.8	17.0	0.35	0.37
2–13	75.5	114	0.39	234	0.36	12.0	17.0	0.35	0.36
2–14	81.7	123	0.40	244	0.36	11.5	17.8	0.36	0.38
2–15	84.5	128	0.43	240	0.36	12.3	17.5	0.34	0.36
2–16	75.5	114	0.38	239	0.36	14.6	17.5	0.30	0.29
2–17	79.5	120	0.41	233	0.35	12.6	17.0	0.33	0.35
2–18	85.5	129	0.45	230	0.35	11.4	16.8	0.36	0.38
2–19	65.4	98.8	0.28	287	...	12.9	20.8	0.30	0.31
2–20	68.4	103	0.28	293	...	12.8	21.4	0.30	0.30
2–22	72.0	109	0.29	299	...	12.5	21.8	0.31	0.31
2–23	81.3	123	0.34	294	...	11.9	21.4	0.32	0.33
2–24	93.5	141	0.39	292	...	10.8	21.3	0.34	0.33
2–25	84.3	127	0.34	298	...	12.7	21.7	0.30	0.30
2–26	90.3	136	0.36	299	...	12.5	21.8	0.31	0.31
2–27	95.7	144	0.39	295	...	11.0	21.5	0.33	0.35
3–8	49.4	83.0	0.44	183	0.38	13.2	13.3	0.37	0.33
3–6	57.7	97.0	0.34	273	0.33	12.5	19.9	0.33	0.33
3–13	63.7	107	0.28	364	...	13.3	26.6	0.28	0.28
4–1	...	...	...	45.0	0.54	17.8	2.2	0.49	0.58
4–2	...	...	...	49.2	0.54	19.9	2.4	0.46	0.50
4–3	...	...	...	74.0	0.49	14.7	3.7	0.48	0.58
4–4	...	...	...	80.2	0.48	16.6	4.0	0.45	0.52
4–5	...	...	...	59.7	0.52	15.4	3.0	0.50	0.60
4–6	...	...	...	61.6	0.52	18.6	3.0	0.45	0.50
4–7	...	...	...	79.2	0.48	8.3	3.2	0.64	0.52
4–8	...	...	...	84.5	0.48	13.0	3.4	0.53	0.39
4–9	...	...	...	61.1	0.51	10.2	2.5	0.64	0.51
4–10	...	...	...	61.6	0.51	13.8	2.4	0.55	0.42
4–11	...	...	...	48.2	0.53	10.0	2.0	0.65	0.54
5–1	...	...	...	115	0.44	21.9	13.2	0.30	0.17
5–2	...	...	...	230	0.35	18.9	26.2	0.22	0.17
5–3	...	...	...	319	...	13.0	36.6	0.26	0.21
6–1	...	...	...	47.6	0.54	9.9	4.4	0.58	0.39
6–2	...	...	...	44.9	0.54	8.7	4.2	0.61	0.45
6–3	...	...	...	43.0	0.55	7.6	4.0	0.65	0.51
6–4	...	...	...	72.2	0.50	7.7	6.8	0.59	0.44
6–5	...	...	...	70.0	0.50	6.8	6.6	0.62	0.47
7–1	...	...	...	257	0.33	11.1	12.5	0.42	0.31
7–2	...	...	...	250	0.33	8.2	12.1	0.48	0.37
7–3	...	...	...	249	0.34	6.9	12.0	0.53	0.40
7–4	...	...	...	532	...	9.2	25.8	0.37	0.27
7–5	...	...	...	523	...	8.2	25.4	0.40	0.28
7–6	...	...	...	523	...	6.9	25.4	0.43	0.31
7–8	...	...	...	815	...	8.2	39.5	0.35	0.23
7–9	...	...	...	845	...	7.3	41.0	0.37	0.25
7–10	...	...	...	276	0.32	21.3	14.5	0.26	0.29
7–11	...	...	...	480	...	19.5	25.2	0.22	0.27
7–12	...	...	...	706	...	20.7	37.2	0.18	0.21
8–1	...	...	...	64.7	0.51	5.3	3.2	0.74	0.55
8–2	...	...	...	68.7	0.50	6.9	3.4	0.68	0.51
8–3	...	...	...	73.5	0.50	8.9	3.7	0.60	0.45
8–4	...	...	...	141	0.42	5.5	7.0	0.65	0.43
8–5	...	...	...	101	0.46	7.3	5.0	0.63	0.43
8–10	...	...	...	145	0.42	5.5	7.3	0.65	0.42
8–11	...	...	...	108	0.46	5.5	5.4	0.70	0.47
8–12	...	...	...	158	0.41	5.5	7.9	0.64	0.42
8–13	...	...	...	213	0.36	5.5	10.6	0.61	0.38
9–6	...	...	...	150	0.41	14.0	9.8	0.39	0.32
9–7	...	...	...	166	0.40	13.0	10.9	0.39	0.32
9–14	...	...	...	86.0	0.48	17.0	5.6	0.40	0.29
9–15	...	...	...	81.8	0.48	15.0	5.4	0.40	0.35
9–16	...	...	...	158	0.47	16.7	10.3	0.40	0.25
9–17	...	...	...	158	0.41	17.0	10.3	0.40	0.25
9–18	...	...	...	194	0.38	15.0	12.7	0.33	0.26
9–19	...	...	...	196	0.38	15.3	12.7	0.33	0.26
10–12	...	...	...	238	0.36	11.5	15.2	0.37	0.37
10–13	...	...	...	182	0.39	12.8	11.6	0.38	0.37
10–14	...	...	...	120	0.45	14.5	7.6	0.41	0.39
10–15	...	...	...	111	0.45	14.9	7.0	0.41	0.38
10–16	...	...	...	105	0.46	18.0	6.5	0.37	0.29
10–17	...	...	...	166	0.41	13.9	10.5	0.37	0.36
10–18	...	...	...	168	0.41	14.9	10.5	0.36	0.33
10–19	...	...	...	170	0.40	13.3	11.4	0.38	0.38
10–20	...	...	...	218	0.37	12.2	13.7	0.38	0.37

[a] Effective furnace radiant heating surface.

Equation [3] has been used, as shown in Table 4, to determine the effective radiant heating surface of the boilers enumerated in the A.S.M.E. Radiation-Committee report.[3] It was deemed best to take F_E equal to unity at the time the calculations were made although a value of 0.95 would perhaps have been preferable.

Data taken from the literature for three additional boiler tests is also included.

Check of Experimental Data by Waterwall Evaporation

Ordinarily the check of experimental data by waterwall evaporation is accomplished by segregation of a part of the furnace waterwalls or, as in the case of the first three units tested in the A.S.M.E. Radiation-Committee report,[3] by constructing the waterwalls so as to have their own feed and steam drum, thus acting as a separate steam-generating unit. Some of the boilers of the Ford Motor Company at Fordson, Mich., are also constructed in this manner. There are several difficulties encountered in the use of this method.

In the first place, results will depend upon the estimated relative heat-absorption rates at different kinds of surfaces. These rates are needed to determine the values of the effective radiant heating surfaces. Two such values are involved. The first is the total effective radiant heating surface in the furnace and the second is that part of the surface for which the absorption was measured. The ratio of the first surface to the second is a factor used in the calculation.

In the second place, certain boilers, as in the case of the first unit of the A.S.M.E. Radiation-Committee report,[3] are constructed so that the furnace-exit slag screen is part of the waterwall boiler. Under these conditions some heat will be transferred by convection through this slag screen into this waterwall boiler.

It is difficult to allow for this heat transfer. In addition, some radiant heat will pass between these slag-screen tubes, be absorbed by the boiler tubes proper and not be measured. In the calculation given in the present paper, these last two quantities of heat have been assumed equal, a very rough approximation.

TABLE 6 DETERMINATION OF μ_L BY HEAT ABSORPTION OF PART OF FURNACE WALLS

1 Name of boiler	3 Firing and fuel	Fraction cold	5 Boiler evap	39 Heat absorbed by screen boiler	40 Heat to furnace based on screen boiler	41 μ_L based on screen boiler evap
			10⁶ Btu per hr			
Cahokia	Pulv. coal	0.25	90	29.4	56.2	0.59
Cahokia	Pulv. coal	0.25	120	29.4	56.2	0.43
Cahokia	Pulv. coal	0.25	144	44.0	84.0	0.52
			1000 kcal per hr			
Sweden	Oil	0.92	727	727	727	0.68
Sweden	Oil	0.92	1257	1257	1257	0.66
Sweden	Oil	0.71	1036	1036	1036	0.61
Sweden	Pulv. coal	0.71	711	711	711	0.57
Sweden	Pulv. peat	0.92	713	713	713	0.61
Sweden	Wood pulp	0.92	446	446	446	.51
			1000 lb per hr	Boiler evap, per cent		
Fordson	Pulv. coal	0.47	150	33	49	0.40
Fordson	Pulv. coal	0.47	300	22	33	0.27
Fordson	Pulv. coal	0.47	450	18	27	0.21

TABLE 7 INDICATED CALCULATIONS FOR ITEMS IN TABLE 5[a]

Item 39..........	[(heat of steam) — (heat of feedwater)] × item 6
Item 40..........	[(rhs[b] in furnace)/(rhs in screen boiler)] × item 39
Item 41..........	(item 40 × item 25)/(item 33 × item 26)
Item 42..........	item 33/rhs
Item 43..........	Fig. 1 using item 42
Item 44..........	calculated using item 20
Item 45..........	item 42/item 25
Item 46..........	Fig. 2 using items 44 and 45
Item 47..........	Equation [2] and Figs. 3 to 11, inclusive.

[a] Items previous to item 39 will be found in Table 3 of the report "An Experimental Investigation of Heat Absorption in Boiler Furnaces."[3]
[b] Effective furnace radiant heating surface.

The plane of the furnace exit (side of boiler tubes adjacent to furnace) is always an effective exposed furnace surface which cannot be segregated for radiant-heat measurement without inducing the problem of convection-heat transfer.

With this in mind, calculations have been made for six boiler furnaces for which test data are available to show the heat absorbed by a portion of the furnace surfaces. Three of these units are boilers Nos. 1, 2, and 3 of the A.S.M.E. Radiation-Committee report[3] for which the computations are given as items 39, 40 and 41 of Table 5 and indicated in Table 7. Table 5 is, in effect, a continuation of Table 3 of the A.S.M.E. Radiation-Committee report.[3] Tests were also available from the literature on boilers for which the construction data of the furnaces are given in Table 4. In this table, boiler No. 11 is a unit at Cahokia Station, St. Louis, Mo. (27); boiler No. 12 is a special test boiler at the Tekniska Hogskolan, Stockholm, Sweden (15, 16); and boiler No. 13 is a Fordson, Mich., unit (10, 19). The μ-determinations, based on segregated furnace-wall surfaces for boilers Nos. 11, 12, and 13 are given in Table 6. The full calculations

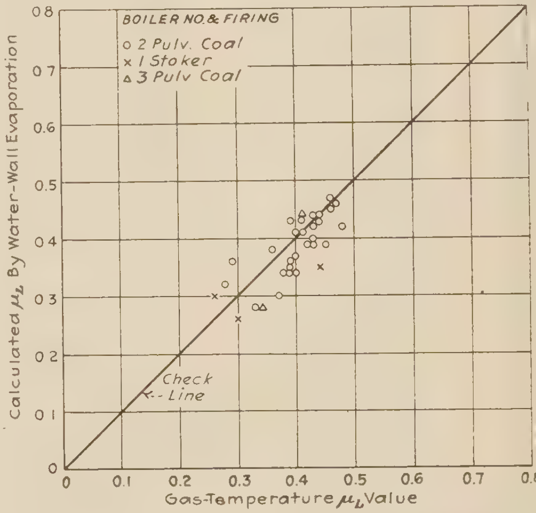

Fig. 13 Comparison of Results by Measurement of Waterwall Evaporation

have been omitted to save space. The μ-values calculated in this manner for boilers Nos. 11, 12, and 13 are compared in Fig. 13 with the values as determined by furnace-exit temperature in the A.S.M.E. Radiation-Committee report[3] for boilers Nos. 1, 2, and 3. Reliable μ-values were not available for the units taken from the literature. The results check approximately. The μ-values, calculated by means of the waterwall-boiler evaporation, are seen to vary somewhat with the flame length, as effected by changes in the primary and tertiary air pressures. Long flames produce a greater heat absorption in the screen boiler than short flames. This causes a variation in the resulting μ-determination.

The gas-temperature μ-values do not exhibit as much variation. It is evident that the increased heat transfer of the surfaces of the waterwall boiler in the case of the long flame would have to be compensated for by decreased heat absorption in the furnace-exit surfaces in order that the gas-temperature μ-values might remain nearly constant.

The foregoing appears logical since long flames occupy more of the bottom portion of the furnace, thus causing increased radiation to the bottom tubes and decreased radiation to the top of the furnace. They also cause considerable convection-heat transfer to the bottom screen-boiler tubes.

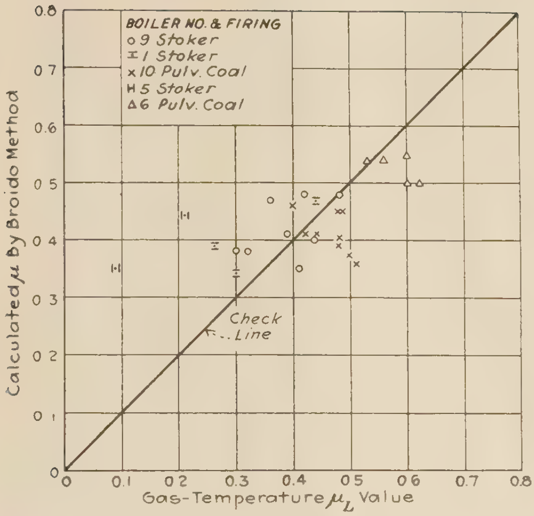

FIG. 14 COMPARISON OF RESULTS BY BROIDO METHOD FOR BOILERS NOS. 1, 5, 6, 9, AND 10[3]

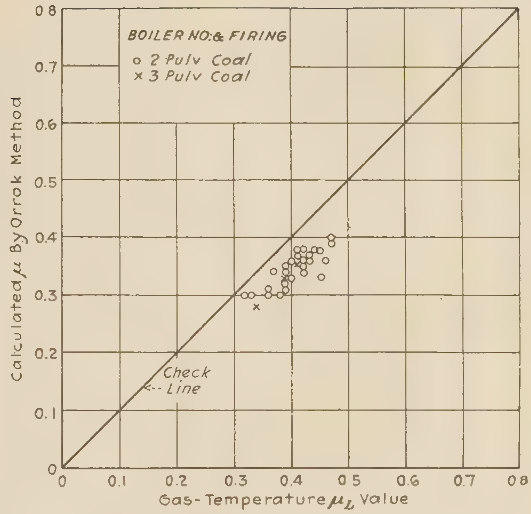

FIG. 15 COMPARISON OF RESULTS BY BROIDO METHOD FOR BOILERS NOS. 2 AND 3[3]

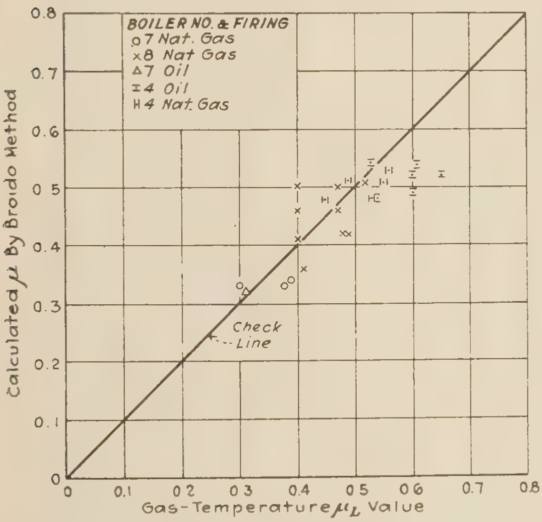

FIG. 16 COMPARISON OF RESULTS BY BROIDO METHOD FOR BOILERS NOS. 4, 7, AND 8[3]

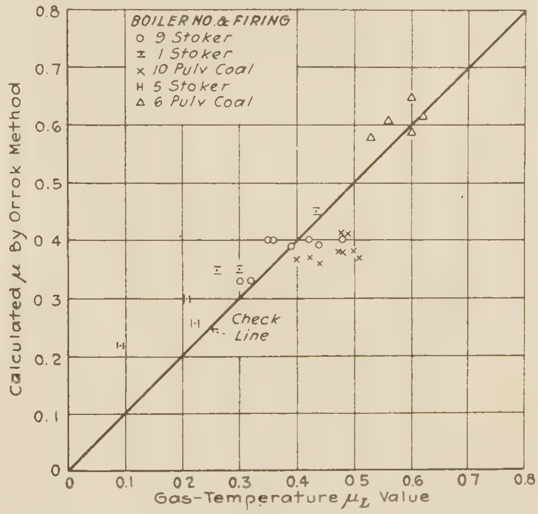

FIG. 17 COMPARISON OF RESULTS BY ORROK METHOD FOR BOILERS NOS. 1, 5, 6, 9 AND 10[3]

It is interesting to note that the values of μ check as closely as they do, but since the method employing the evaporation of the waterwall boiler is limited to special cases, and the gas-temperature μ-values determined by the method of furnace heat balance are more accurate, those determined by the evaporation method appear to be of no particular importance.

CHECK OF BROIDO, ORROK, AND WOHLENBERG METHODS

In Table 5 are given calculations for the tests on the units for which data is given in the A.S.M.E. Radiation-Committee report.[3] Item 43 in the table gives the μ-values obtained by the Broido method, item 46 the μ-values obtained by the Orrok method, and item 47 the μ-values obtained by the Wohlenberg method.

The results obtained by these three methods are compared in Figs. 14 to 21, inclusive, to the gas-temperature μ-values as given by A.S.M.E. Radiation-Committee report.[3] The line marked "check line" in these figures shows where the points lie if the two values check each other. The boiler numbers given in the various charts and tables correspond with those of the A.S.M.E. Radiation-Committee report.[3]

It should be again emphasized that in any usage of the Wohlenberg, Broido, or Orrok formulas, the results are dependent on the method of calculating the effective furnace radiant heating surface and that the surface calculated in a manner different from that used in this paper would yield different results.

It will be noted that the Broido determinations are surprisingly close considering the fact that only a single curve is used in the method.

The Orrok results are fairly representative except for the natural-gas tests for which the formula yields rather high values. The Orrok formula was probably not intended for natural-gas fuels. Further comment on these findings does not seem necessary, as the relative value of the methods, for the units for which tests are available, is quite apparent from the curves.

In view of the foregoing results it appears on the surface as though the values for gas radiation given by Schack (59) and used in the Wohlenberg method are somewhat low.

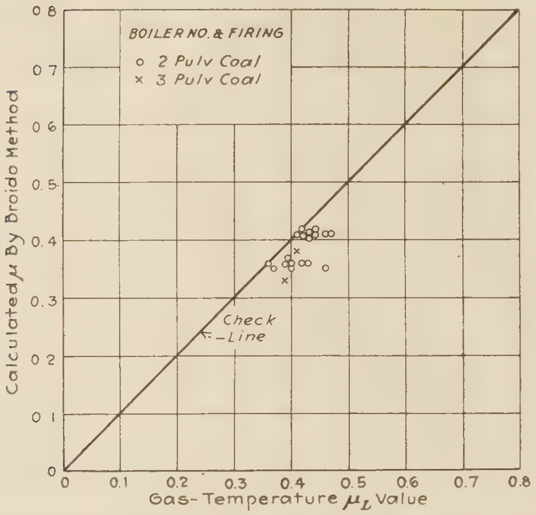

Fig. 18 Comparison of Results by Orrok Method for Boilers Nos. 2 and 3[3]

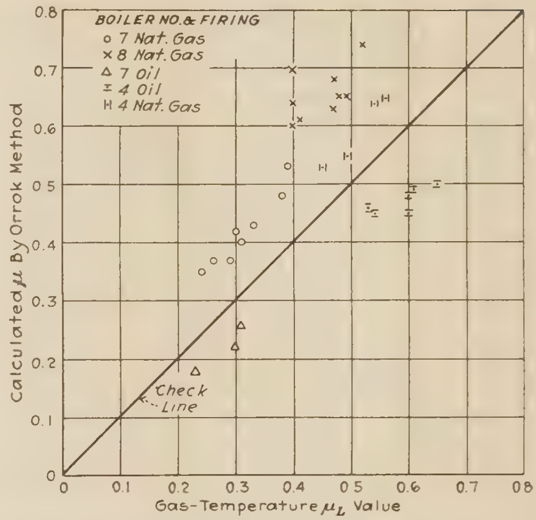

Fig. 19 Comparison of Results by Orrok Method for Boilers Nos. 4, 7, and 8[3]

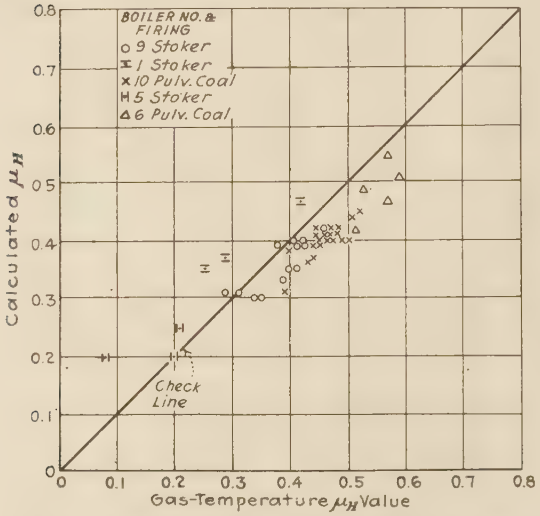

Fig. 20 Check of Wohlenberg Method for Boilers Nos. 1, 5, 6, 9, and 10[3], Using the New Coefficients (F-Factors)

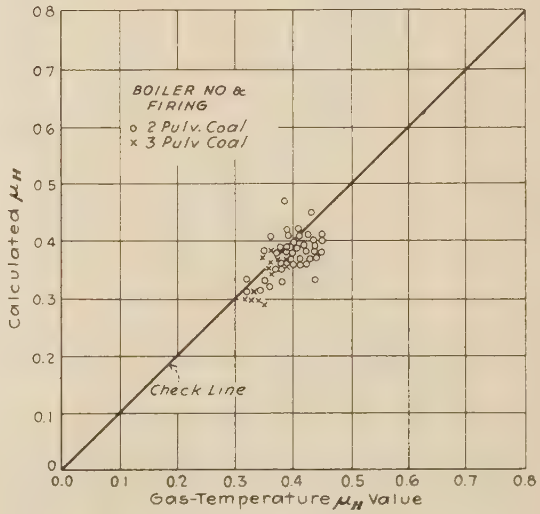

Fig. 21 Check of Wohlenberg Method for Boilers Nos. 2 and 3[3], Using the New Coefficients (F-Factors)

This apparent discrepancy may, however, be due to the fundamental assumption of the Wohlenberg method that the furnace-exit-gas temperature is also equal to the mean radiating furnace temperature.

The so-called straight Stefan-Boltzman law method has had so many different values given for the constants therein contained, and is in such a crude form as ordinarily used, that no determinations have been made by this method for presentation in this report. While the Stefan-Boltzman law is used in the Wohlenberg derivation, it is only one of the factors used to obtain a solution, and therefore the method is not classified under the foregoing heading. An attack using the Stefan-Boltzman law is included in a paper by one of the present authors.[6]

An Experimental Investigation of Heat Absorption in Boiler Furnaces

By W. J. Wohlenberg,[1] New Haven, Conn., H. F. Mullikin,[2] New York, N. Y., W. H. Armacost,[1] New York, N. Y., and C. W. Gordon,[3] Chicago, Ill.

This paper is a report of the A.S.M.E. Committee on Absorption of Radiant Heat in Boiler Furnaces and includes test data obtained on ten boilers in different power plants in the United States. It involves stoker-fired, pulverized-fuel, and oil- and gas-fired units with various combinations of refractory and water-cooled furnaces. The data include furnace-exit gas temperatures as measured by means of the high-velocity thermocouple. These temperatures are used to determine the fraction of heat input to the furnace that was absorbed in its walls.

At THE time the committee[4] submitting this report was formed it was suggested that radiation tests on large commercial boiler furnaces be made to supplement work of a more fundamental nature which was being conducted by various agencies. As a result a fellowship, sponsored by the Superheater Company, was established under the joint direction of Professor Wohlenberg of Yale University and Messrs. R. M. Gates, R. M. Osterman, and C. W. Gordon to study the problem and make such tests as might be necessary to put the subject on a more practical basis. H. F. Mullikin was appointed research fellow under this arrangement and had charge of the tests.[5] These tests were made on ten boilers in different power plants in the United States and involve stoker, pulverized-fuel, oil, and gas firing with various combinations of refractory and water-cooled furnaces.

Before describing these tests in detail, it seems advisable to discuss briefly the more important papers which have appeared on the subject. It is not proposed to outline in detail the previous material printed on this subject but merely to refer to the works of past authors.

Since the general adoption of the waterwall furnace for large steam-generating units beginning about 1924, many articles have been written offering various methods of obtaining a solution to the furnace heat-transfer problem. The majority of these have been technical, and are based on physical constants and laws, and on observed general conditions in furnaces. Several have been based on empirical test data.

The principal objection to any type of solution based on theory is that it is difficult to take into account a large number of design variables such as type of fuel; fineness of fuel; location and type of burners; location, type and surface of waterwalls and exposed boiler tubes; volume and shape of furnace; combustion characteristics of different fuels or of the same fuel under slightly different conditions; and such operating factors as slag on surfaces, flame length, draft, flame impingement, flame volume, excess air, method of air supply, and temperature of air.

Because of the difficulties involved in making heat-transfer tests, very few of them have been made, and it is upon the data from this insufficient number of tests that most empirical methods for solving heat-transfer problems are based. Therefore, it is questionable as to what extent these results can be applied. The empirical formulas generally neglect most of the factors and depend only upon one or two of the principal variables. Very few of the methods have not been checked by any one other than the original investigator.

With these facts in mind, attention is called in the bibliography to some of the more notable attacks on the problem. The list is not claimed to be complete but it is believed that most of the methods yielding fairly reasonable quantitative results are included.

In 1925 Broido (7)[6] presented a paper giving an excellent general discussion of the problem with a single curve to summarize several tests. Wohlenberg and various associates have presented numerous papers (1, 2, 3, 4, 5) containing an attack on the problem from a fundamental theoretical basis. Modified forms of this method, some of which are better adapted to practical application, are given by Haslam and Hottel (13, 60), and Münzinger (21, 41, 55). Even so, it is difficult to evaluate certain practical conditions in the process of solution. Orrok (20) has presented a formula empirically derived from Hudson (39). Wilson, Lobo, and Hottel (45) have adapted the Orrok equation in oil-still design. In most of these methods the Stefan-Boltzman law has been adapted.

Others making use of the Stefan-Boltzman law are: Schack (6, 59), Roszak and Veron (23), Gerbel (40), Ritchie (42), Forssblad (9), Seibert (48), and Müller (46). Rosin and Fehling (47) have developed theoretical equations to give maximum furnace heat liberation rates. DeBaufre (17) has reported tests on several units and has used a furnace heat balance to correlate his results. Kuhn (14) and Friedrich (54) have reported tests similar to those of DeBaufre.

A large number of tests have been run on boiler furnaces in which one or more tubes were isolated and the actual heat absorbed was measured by water circulated through the tubes. Results have in most cases been reported, but no attempt was made to derive an equation or formula therefrom. A number of these tests have been reported by Broido and Orrok (7). Others may be found in some of the N.E.L.A. serials (27, 28, 29, 33), transactions of various societies (10, 25, 34, 36, 37) and other sources (19, 26). A number were reported in papers by Artsay (12) and Ramzin (22).

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[2] Jun. A.S.M.E.

[3] Assoc-Mem. A.S.M.E.

[4] The Committee on Absorption of Radiant Heat in Boiler Furnaces consists of W. J. Wohlenberg, Chairman, E. G. Bailey, R. M. Gates, C. W. Gordon, E. L. Lindseth, Geo. A. Orrok, R. J. S. Pigott, and John Van Brunt.

[5] This report abstracts in part a dissertation presented by Mr. Mullikin for the degree of Doctor of Philosophy at Yale University.

Contributed by the Special Research Committee on Absorption of Radiant Heat in Boiler Furnaces for presentation at the Annual Meeting December 2 to 6, 1935, of The American Society of Mechanical Engineers.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1936, for publication at a later date.

[6] Numbers in parentheses correspond to similarly numbered references given in the bibliography at the end of the paper.

TABLE 1 GENERAL BOILER DESCRIPTION AND DATA

General data:											
1	Boiler number	1	2	3	4	5	6	7	8	9	10
2	Rated surface, sq ft	24,600	27,548	27,548	10,830	15,000	10,760	15,150	15,930	22,900	60,706
3	Type boiler	S[a]	DS[b]	DS	S	HX[c]	HX	HX	HX	HX	DS
4	Make boiler	B&W[d]	W&W[e]	B&W	Heine	CE[f]	Spring[g]	W&W	W&W	Spring	CE
5	Min steam rating, 1000 lb per hr	100	....	....	....	....	40	....	60	....	....
6	Normal steam rating, 1000 lb per hr	225	225	225	120	....	....	120	175	....	600
7	Max steam rating, 1000 lb per hr	340	400	400	160	....	200	230	325	....	1250
8	Date in service	1929	1929	1924	1931	1931	1929	1930	1929	1928	1929
9	Rated preheater surface, sq ft	26,930	8225	2425	....	13,660	11,550	none	none	....	82,721
10	Make preheater	B&W	F-W[h]	Special	Ljun[i]	F-W	Thermix	....	....	Ljun.	CE[f]
11	Calc. furnace volume to exit, cu ft	14,000	22,410	22,410	11,000	5420	8740	5550	13,200	9040	33,400
12	Add. furnace volume above exit, cu ft	....	1930	1930	....	....	....	....	....	....	4040
13	Furnace width, ft	26.270	20.062	23.062	16.00	24.166	18.917	23.00	23.791	27.75	23.166
14	Make furnace	FL[j]	CE[f]	B&W	CE	FL	Spring[g]	....	CE	TS[k]	CE
15	Burners per side	....	8	8	8	....	2	14	14	....	20
16	No. sides fired	....	2	2	1	....	1	1	2	....	2
17	Type burner	....	Fish[l]	Fish	....	....	HT[m]	....	....	....	Fish
18	Make burners	....	Lop[n]	Lop	....	....	P[o]	F[p]	F	....	Lop
19	Primary air, %	100[q]	15	15	100[q]	100	30[q]	100	100	100[q]	25[q]
20	Secondary air, %	....	50	50	....	....	70[q]	....	....	....	75[q]
21	Tertiary air, %	....	35[q]	35[q]	....	....	....	....	....	....	....
22	Type stoker	UF[r]	....	....	....	CG[s]	....	....	....	UF	....
23	Make stoker	Taylor	....	....	....	Ill.	....	....	....	Riley	....
24	Area of stoker, sq ft	605	....	....	....	528	....	....	....	360	....
25	Retorts, stoker	15	....	....	....	....	....	....	....	15	....
26	Tuyères, stoker	57	....	....	....	....	....	....	....	22	....
27	Length, stoker	19.2	....	....	....	22	....	....	....	13	....
Coal data:											
28	Moisture, %	2.70	2.45	2.45	....	21.50	9.11	....	....	4.10	0.25
29	Volatile, %	34.18	35.55	35.55	....	26.50	37.63	....	....	19.43	19.85
30	Fixed carbon, %	56.40	54.73	54.73	....	32.00	34.75	....	....	68.70	72.42
31	Ash, %	6.72	7.27	7.27	....	20.00	18.51	....	....	7.77	7.48
32	Ash fusion, deg F	....	....	....	....	....	....	....	....	2230	....
33	Oxygen, %	5.93	8.65	8.65	....	5.88	15.88	....	....	2.68	4.38
34	Sulphur, %	0.92	0.86	0.86	....	6.07	5.98	....	....	1.80	1.36
35	Ash, %	6.90	7.45	7.45	....	25.40	17.90	....	....	8.07	8.02
36	Nitrogen, %	1.23	1.49	1.49	....	0.92	1.10	....	....	1.41	1.25
37	Hydrogen, %	5.09	5.20	5.20	....	3.75	4.57	....	....	4.53	4.29
38	Carbon, %	79.93	76.35	76.35	....	58.00	54.57	....	....	81.51	80.70
39	High heat. value, dry, Btu per lb	14,230	13,800	13,800	....	10,500	11,020	....	....	14,509	14,580
40	High heat. value, as fired, Btu per lb	13,850	13,450	13,450	....	8250	10,050	....	....	14,150	14,550
41	Through 200 mesh, %	....	55.7	55.7	....	....	47.2	....	....	....	....
42	Through 100 mesh, %	....	75.6	75.6	....	....	62.4	....	....	....	....
43	Through 50 mesh, %	....	91.6	91.6	....	....	....	....	....		....
44	Through 30 mesh, %	....	98.2	98.2	....	....	92.6	....	....	....	....
Oil data:											
45	Carbon, %	....	....	....	89.8	....	....	....	....	....	....
46	Hydrogen, %	....	....	....	8.95	....	....	....	....		....
47	Oxygen, %	....	....	....	0	....	....	....			....
48	Sulphur, %	....	....	....	0.93	....	....	0.75	....	....	....
49	Nitrogen, %	....	....	....	0	....	....	....	....	....	....
50	Specific gr at 60 F	....	....	....	1.036	....	....	0.96	....	....	....
51	High heat. value, as fired, Btu per lb	....	....	....	18,424	....	....	19,000	....	....	....
52	Low heat. value, as fired, Btu per lb	....	....	....	17,400	....	....	17,970	....	....	....
53	Flash point, F	....	....	....	355	....	....	275	....	....	....
Natural-gas data											
54	Ethane, %	....	....	....	1.70	....	....	1.28	7.93	....	....
55	Methane, %	....	....	....	95.50	....	....	90.38	81.90	....	....
56	Oxygen, %	....	....	....	0.20	....	....	0.11	0.34		....
57	Carbon dioxide, %	....	....	....	2.00	....	....	0.22	....	....	....
58	Nitrogen, %	....	....	....	....	....	....	8.01	9.77	....	....
59	Specific gravity	....	....	....	0.636	....	....	....	....	....	....
60	High heat. value at 30-in. hg, and 60 F, Btu per cu ft	....	....	....	993	....	....	....	970	....	....
61	Low heat. value at 30-in. hg, and 60 F, Btu per cu ft	....	....	....	903	....	....	....	....	....	....
Calculated refractory surface:											
62	Front or uptake, sq ft	....	776	776	340	480	62	347	265	....	92
63	Each side, sq ft	....	523	750	85	28	....	241	157	....	....
64	Rear or downtake, sq ft	....	776	776	76	600	80	310	193	83	92
65	Top, sq ft	....	387	387	....	....	....	....	....	....	....
66	Bottom, sq ft	100	....	....	314	....	....	364	452	97	....
Refractory-faced waterwall block:											
67	Front, surface, sq ft	446		....		....					
68	Front, amt slagged, %	75				....					
69	Front, slag thickness, in	0.75				....					
70	Each side, surface, sq ft	....				81					....
71	Each side, amt slagged, %	....				50		....			
72	Each side, slag, thickness, in	....	....	....		0.75					....
73	Top, surface, sq ft	170		....							....
74	Top, amt slagged, %	75	....								
75	Top, slag thickness, in	0.75	....				....				
76	Bottom, surface, sq ft	....	....	....		....	417	....			....
77	Bottom, amt slagged, %	....					100	....			
78	Bottom, slag thickness, in	....	....		....		0.125	....			....
Bare-faced waterwall block:											
79	Front surface, sq ft	....	....							83	353
80	Front, amt slagged, %	....								100	25
81	Front, slag thickness, in	....		....		....				0.125	0.25
82	Each side, surface, sq ft	244	....	....		117				39	....
83	Each side, amt slagged, %	20	....			50				100	....
84	Each side, slag thick, in	0.75	....			0.75				0.125	....
85	Rear, surface, sq ft	278	....	....						83	353
86	Rear, amt slagged, %	80	....	....				....		100	25
87	Rear, slag thick., in	0.75	....	....				....		0.125	0.25

TABLE 1 (*Continued*)

Bare-tube waterwall:											
88	Front, diam tubes, in.	....	....	....	....	....	3.0	....	....	3.5	....
89	Front, distance center to center, in.	....	....	....	....	....	3.5	....	....	$10\frac{5}{8}/2$	....
90	Front, total surface, sq ft	....	....	....	....	....	407	....	....	417	....
91	Front, amt slagged, %	....	....	....	....	....	100	....	....	100	....
92	Front, slag thickness, in.	....	....	....	....	....	0.125	....	....	0.125	....
93	Each side, diam tubes, in.	3.25	3.25	....	3.00	....	3.00	....	....	3.50	....
94	Each side, distance center to center, in.	6.00	9.00	....	10.25	....	3.50	....	....	8.5/2	....
95	Each side, total surface, sq ft	290	227	....	180	....	449	....	....	287	....
96	Each side, amt slagged, %	55	30	....	....	....	100	....	....	100	....
97	Each side, slag thickness, in.	0.75	0.75	....	....	....	0.125	....	....	0.125	.
98	Rear, diam tubes, in.	3.25	....	....	3.00	....	3.00	....	....	3.50	.
99	Rear, distance center to center, in.	6.0	....	....	5.125	....	3.50	....	....	8.5/2	.
100	Rear, total surface, sq ft	384	....	....	364	....	294	....	....	333	
101	Rear, amt slagged, %	50	....	....	....	....	100	....	....	100	....
102	Rear, slag thickness, in.	0.75	....	....	....	....	0.125	....	....	0.125	....
103	Bottom, diam tubes, in.	....	3.25	3.25	....	....	....	....	....	....	4.0
104	Bottom, distance center to center, in.	....	9.0	9.0	....	....	....	....	....	....	10.5
105	Bottom, total surface, sq ft	....	860	860	....	....	57	....	....	....	945
106	Bottom, amt slagged, %	....	....	....	....	....	....	....	....	....	25
107	Bottom, slag thickness, in.	....	....	....	....	....	....	....	....	....	0.25
Fin-tube waterwall:											
108	Front, diam tubes, in.	....	....	....	....	....	....	....	4.0	....	4.0
109	Front, distance center to center, in.	....	....	....	....	....	....	....	7.0	....	10.5
110	Front, total surface, sq ft	....	....	....	....	....	....	....	486	....	555
111	Front, amt slagged, %	....	....	....	....	....	....	....	....	....	25
112	Front, slag thickness, in.	....	....	....	....	....	....	....	....	....	0.25
113	Each side, diam tubes, in.	....	....	....	3.0	....	....	....	4.0	....	4.0
114	Each side, distance center to center, in.	....	....	....	5.125	....	....	....	7.0	....	6.0
115	Each side, total surface, sq ft	....	....	....	400	....	....	....	400	....	1447
116	Each side, amt slagged, %	....	....	....	....	....	....	....	....	....	25
117	Each side, slag thickness, in.	....	....	....	....	....	....	....	....	....	0.25
118	Rear, diam tubes, in.	....	....	....	3.0	....	....	....	4.0	....	4.0
119	Rear, distance center to center, in.	....	....	....	5.125	....	....	....	7.0	....	10.5
120	Rear, total surface, sq ft	....	....	....	355	....	....	....	306	....	555
121	Rear, amt slagged, %	....	....	....	....	....	....	....	....	....	25
122	Rear, slag thickness, in.	....	....	....	....	....	....	....	....	....	0.25
123	Top, diam tubes, in.	....	....	....	....	....	....	....	....	....	4.0
124	Top, distance center to center, in.	....	....	....	....	....	....	....	....	....	10.5
125	Top, total surface, sq ft	....	....	....	....	....	....	....	....	....	508
126	Top, amt slagged, %	....	....	....	....	....	....	....	....	....	25
127	Top, slag thickness, in.	....	....	....	....	....	....	....	....	....	0.25
Furnace exit:											
128	Furnace exit, diam tubes, in.	3.25	3.25	3.25	3.0	4.0	3.0	4.0	4.0	3.0	3.0
129	Furnace exit, effective length, ft	15.00	17.75	....	22.0	17.0	19.0	15.5	20.0	20.0	18.66
130	Effective length of tubes, ft	....	20.00	20.0	....	....	....	....	....	....	29.0
131	Furnace exit, effective width, ft	26.25	24.75	....	16.50	24.00	18.92	23.00	23.75	27.75	23.10
132	Distance center to center, first row, in.	12.0	....	6.0	....	14.0	10.625	16.25	7.0	10.75	16.0
133	Second row tubes	sg[t]	sg	sg	pll[u]	sg	pll	sg	sg	pll	sg
134	Distance between first and second rows, in.	9.0	6.0	6.0	6.5	12.0	4.0	8.125	7.0	4.0	7.0
135	Distance, center to center second row, in.	12.0	6.0	6.0	6.0	14.0	10.625	16.25	7.0	10.75	16.0

[a] Stirling; [b] Double Stirling; [c] Horizontal cross-drum; [d] Babcock & Wilcox; [e] Walsh & Weidner; [f] Combustion Engineering; [g] Springfield; [h] Foster-Wheeler; [i] Ljüngstrom; [j] Fuller-Lehigh; [k] Superheater Company; [l] Fishtail; [m] horizontal turbulent; [n] Lopulco; [o] Peabody; [p] Forney; [q] preheated air; [r] underfeed; [s] chain grate; [t] staggered; [u] parallel.

Bailey (24) has developed a method of measuring heat absorption in furnaces having waterwalls covered with refractory facings. Schack has suggested the use of the radiation pyrometer for such measurements (6, 13). Koessler has used the thermopile (32). Croft and Schmarje (56) have used an absorption calorimeter to measure incident radiation to the walls of a furnace.

In Sweden, Lindmark and his associates have reported results (15, 16, 43) of tests involving the use of oil and pulverized fuel in an experimental furnace. In this country Sherman has published (51, 52, 53) considerable material on an experimental installation burning pulverized coal and gas. Brown advanced an empirical curve (8) which Marshall has discussed (44). Bottomley has presented a paper (49) based on data reported by Orrok.

Sherman has reported data on furnaces (31) taken during the course of an investigation on refractories. Artsay has presented two papers (11, 12) giving an analysis based on hydraulic similitude. Data on gas radiation have been reported by numerous investigators who will not be mentioned here.

Methods for Determining Radiant-Heat Absorption

Consideration of the previously described articles and papers reveals that five principal methods have been used in attempts to determine radiant-heat absorption in boiler furnaces. They are (*a*) determination of the actual evaporation in one or more tubes exposed to radiant heat, (*b*) use of physical methods such as the thermopile or radiation calorimeter, (*c*) the establishment of a temperature gradient through conducting material (only applicable to refractory-faced blocks), (*d*) a determination of the furnace temperature and subtraction of the heat leaving the furnace in the gases from the heat input to the furnace, and (*e*) the use of known physical data such as the Stefan-Boltzman law, and values of gas radiation.

The first four of these methods are empirical whereas the last is primarily a theoretical attack. Obviously the ideal procedure would be to develop first an attack which would give theoretical formulas to cover all cases and conditions, and then second to determine with the aid of data supplied from tests, the values of constants which appear in the theoretical formulas.

Unfortunately, the data given in the literature for each individual test usually are so meager that it is impossible to coordinate data from various sources in order to show the effect of such dimensions as furnace volume, and radiant heating surface. Thus, the committee submitting this report decided to conduct tests on a number of existing furnaces in order to obtain data which could be used as a reference for checking theoretical or empirical formulas.

The first of the four previously mentioned practical methods for determining radiant-heat absorption (segregating portions of the waterwall) is not ordinarily feasible in the case of the usual boiler furnace because of the expense and time involved. There are perhaps half a dozen boiler installations in the United States where the waterwall circulation is in some manner separated entirely from the boiler proper. Tests on three of these installations are given in this paper. Another notable example is the boiler installation at the Ford Motor Co., Fordson, Michigan, (10, 19), in which the side-wall tubes constitute an entirely separate boiler. Even here the top tubes of the furnace and the lower

boiler-tube row belong to the boiler proper and hence their heat absorption cannot be experimentally determined in this manner.

During the preliminary work on tests conducted by the committee, attempts were made to use a radiation calorimeter or "thermaprobe" as it has been termed. It was made of a flat-faced hollow steel block through which water was circulated. The heat absorbed by the circulating water was found by measuring its weight per unit of time, and its initial and final temperature. This method was discarded as being unsatisfactory. The objections are: (1) Limitation as to number and location of openings which may be made in the furnace walls, (2) difficulty in controlling the water flow at a constant value, (3) difficulty of operating more than one thermaprobe at the same time, (4) lag in responsivity during time intervals in which furnace conditions change, (5) difficulty in obtaining the same amount of slag on the thermaprobe that existed on the furnace tubes, or of allowing for it by correction and (6) possibility of considerable error due to peculiar local conditions.

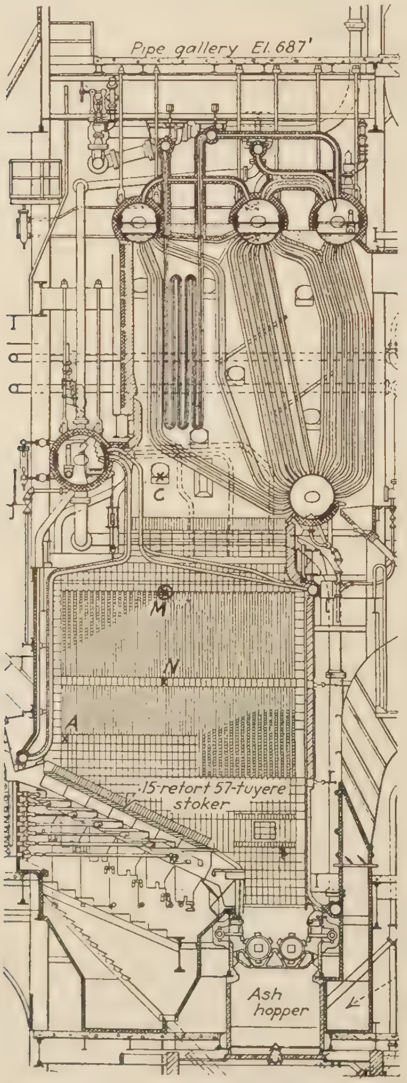

Fig. 1 Underfeed Stoker-Fired Boiler With Waterwalls on All Four Sides and Separate Waterwall Boiler Circulation

TABLE 2 DATA FROM SPECIAL TEST ON UNIT SHOWN IN FIG. 1

Boiler evap, lb per hr	Temperatures, deg F				
	Point A	Point N	Point M	Point C^a	Point C^b
115,000	2455	2125	2150	1940	1985
215,000	2920	2525	2630	..	2215
300,000	2780	2810	2810	2490	2505

[a] North side of boiler.
[b] South side of boiler.

The third method for determining radiant-heat absorption given by Bailey (24), is not applicable to bare-tube or fin-tube construction.

It appeared that the fourth method, which involves measuring the furnace-outlet gas temperature and setting up a furnace heat balance along the lines suggested by Kuhn (14) and DeBaufre (17), furnishes the most feasible method for determining heat transfer in boiler furnaces. The difference between the test procedures given by Kuhn and DeBaufre lay in the method of measuring gas temperatures. Kuhn used a high-velocity thermocouple, which is at present generally conceded to be the most accurate method of temperature measurement. DeBaufre, it is understood, used ordinary bare thermocouples.

Test Apparatus

The high-velocity thermocouple used in making the tests discussed in this report is shown in Fig. 11. The instrument consists of a water-cooled brass tube which can be inserted into the gas stream, the usual distance being 92 in. minus the wall thickness. The gases are aspirated past the hot junction of the thermocouple at a very high velocity by means of the air ejector. This increases the heat transfer from the gas to the thermocouple, thereby causing the thermocouple to read close to the true gas temperature. A more complete theory and description has been given elsewhere (18).

For some of the tests a radiation pyrometer was available which was used to take readings, where possible, for use according to Schack's (6) or Koessler's (32) method. These readings were later discarded, as early experiments showed that only indefinable mean results could be obtained by use of the radiation pyrometer. However, they did show that under certain conditions the emissivity of the furnaces, as indicated by a comparison of the high-velocity thermocouple and radiation-pyrometer determinations, may approach close to the value of unity.

Description of the Boilers Used in the Tests

In choosing boilers for test purposes, the committee endeavored to select representative boilers of various types burning various fuels in different manners. The following boilers were selected for testing:

1 Semivertical underfeed stoker-fired boiler, Fig. 1, with waterwalls on all four sides and 24,600 sq ft of heating surface.

2 Double-ended pulverized-coal-fired boiler, Fig. 2, with waterwalls on two sides, a slag screen on the bottom, and 27,548 sq ft of heating surface.

3 Double-ended pulverized-coal-fired boiler, Fig. 3, with a slag screen on the bottom and 27,548 sq ft of heating surface.

4 Semivertical combination gas- and oil-fired boiler, Fig. 4, with water walls on four sides and 10,830 sq ft of heating surface.

5 Horizontal cross-drum boiler, Fig. 5, with waterwalls on two sides, 15,000 sq ft of heating surface, and equipped with a chain-grate stoker.

6 Horizontal cross-drum boiler, Fig. 6, water cooled on four sides and the bottom. This boiler has 10,760 sq ft of heating surface and burns pulverized fuel.

7 Horizontal cross-drum combination gas- and oil-fired boiler, Fig. 7, with a refractory furnace and 15,150 sq ft of heating surface.

8 Horizontal cross-drum gas-fired boiler, Fig. 8, with waterwalls on four sides and 15,930 sq ft of heating surface.

9 Horizontal cross-drum boiler, Fig. 9, with waterwalls on four sides and 22,900 sq ft of heating surface. This boiler is equipped with an underfeed stoker.

10 Double-ended boiler, Fig. 10, with waterwalls on all sides and 60,706 sq ft of heating surface. This unit burns pulverized fuel.

Thus, data in this report include those from tests on three stoker-fired boilers, three gas-fired boilers, two oil-fired boilers, and four pulverized-coal-fired boilers.

A detailed description of these boilers is given in Table 1. This table also includes the various carefully measured exposed radiant surfaces, as well as construction and firing data. A considerable amount of data are given in order to furnish complete information concerning the tests, as it is not known exactly what factors influence the results obtained.

The surface areas given in Table 1 are overall values measured on the particular furnace walls taken as planes. The proximate analyses given are usually based on the fuel used during the tests. It was not always possible to take an ultimate analysis, and therefore, in some cases the values for the ultimate analyses are for the average station fuel rather than the specific values for the coal used during the tests. An estimate of the slag on the actual waterwall surfaces has been made. It was impossible to judge flame volumes to any degree of accuracy.

Preliminary Tests

Since the temperature measured at the furnace outlet (denoted by M in the figures) was the most important determination of the tests it is advisable to note some of the preliminary work.

During the tests on the first unit, Fig. 1, it was decided to measure temperatures by high-velocity thermocouples on one side of the boiler at points A, M, N, and C, and on the other side of the boiler at point C. Some of the data obtained at these points on the boiler shown in Fig. 1 are given in Table 2.

The temperature differences at C between the two sides of the boiler are less than 50 F at low loads and less than 20 F at high loads. The high-velocity thermocouples could be inserted up to a maximum distance of 7 ft into the boiler pass. Since this boiler, Fig. 1, is 24 ft wide, this left a space of 12 ft in the middle (less in the other boilers) which could not be reached by the high-velocity thermocouples. However, it was noted early and confirmed repeatedly that the gas temperature within reach of the tip of the high-velocity thermocouple showed no discernible temperature gradient to within 20 in. of the wall. On this account it is believed that the temperatures in the middle of the furnace width are not appreciably higher than at a point 6 or 7 ft from the wall.

Moreover, the furnace-gas temperatures are subject to minute-to-minute variations in the neighborhood of 100 F in most of the boilers tested, especially in the stoker-fired units.

Thus, it would seem that one reading on one side of the furnace width, while not as desirable as an average of temperatures taken across the pass and weighted according to mass flow, is nevertheless, over a period of time quite indicative of the average furnace-exit temperature within the accuracy of the other factors governing the tests.

The maximum error caused by this procedure of measuring temperatures will obtain at low loads where stratification of gas is more apt to occur. At high loads the gases will fill the furnace more uniformly and will promote temperature conditions which are more nearly uniform. Likewise, the furnace temperatures will be higher and since radiation is roughly proportional to the fourth power of the temperature, this radiation will also tend to equalize gas temperatures in various parts of the furnace cavity. This last statement is confirmed by the data of Table 2 where it will be seen that the temperatures at A, N, and M are in close agreement at the high load. Possibly it might be considered that high-load tests yield more accurate results than tests at lower loads.

It is possible that the temperature gradient frequently observed across boiler passes in the neighborhood of the superheater is more pronounced than in the hotter zones. This is due to unequal convection-heat absorption and the fact that radiation is insufficient to permit temperature equalization.

Description of Tests and Test Data

The boilers tested are shown in Figs. 1 to 10, inclusive, and are numbered from 1 to 10, respectively, in the tables. The furnace-exit temperature is taken at the point M as shown in the figures. This point was, whenever possible, chosen so as to be in the main gas stream approximately 1 ft underneath the boiler tubes at the furnace exit and about 6 ft toward the center of the furnace, measured from the inside furnace wall. Measurements were taken on one side of the furnace. The steam flows were measured by flowmeters which were usually "water-columned" before the tests. The flowmeter readings are believed to be within 5 per cent of the true values, which is within the accuracy desired. Flowmeter corrections due to pressure and temperature were not made as they were small. The boiler-outlet temperatures were averages determined by a number of bare thermocouples or by calibrated recorders. Steam pressures were obtained by checked station gages. Feedwater and steam temperatures were obtained by thermometers or thermocouples.

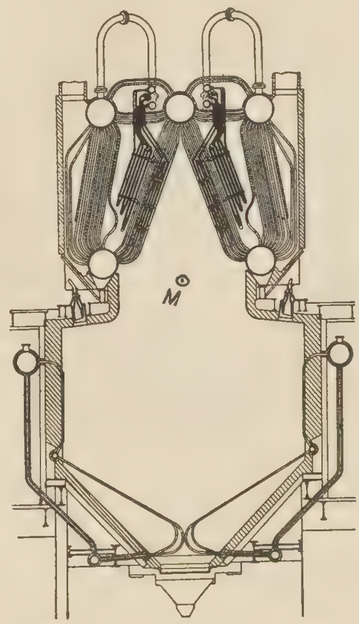

Fig. 2 Double-Ended Pulverized-Coal-Fired Boiler With Separate Waterwall-Boiler Circulation and Two Side Walls of Furnace Water Cooled

Various auxiliary data which may be of interest were also included. The gas analysis is an average for samples taken across the top of the first pass or at the boiler outlet as noted in item 16 of Table 3.

The thermoelements of the high-velocity thermocouples were calibrated before a number of the tests and were always found to be within 10 F of their original calibration. The gas temperatures within the furnace, within the reach of the end of a high-velocity thermocouple located at any opening, were nearly always found to be constant at all points up to within about 2 ft of the

inside wall of the furnace for any of the tests, neglecting momentary total furnace-temperature variations of as much as 100 F.

A number of the readings of the high-velocity thermocouple were taken in various portions of the furnace and in the boiler passes but, as this paper is confined to the calculation of furnace radiation, they have not been included.

The furnace-temperature readings are usually averages of separate readings taken 10 minutes apart, the high-velocity thermocouple being removed between readings to avoid slagging difficulties. Duration of tests varied from 30 minutes to $3^1/_2$

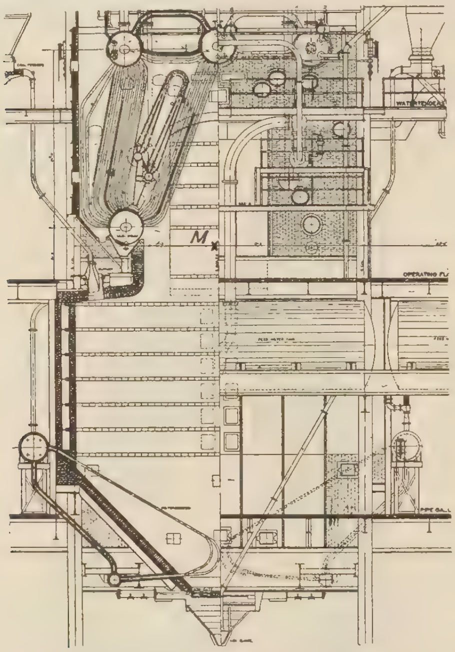

Fig. 3 Double-Ended Pulverized-Coal-Fired Boiler With Separate Waterwall-Boiler Circulation and Water Cooling Only on Bottom of Furnace

hours according to conditions governing boiler operation at the individual power plants.

The percentage of moisture in the steam entering the superheater was measured in almost all of the tests and was found to be less than 0.5 per cent.

The data of a number of specific tests are given in Table 3 in some detail. Due to space limitations a large number of the less important tests are not included.

The first boiler which was tested is fired by an underfeed stoker and has a separate screen or waterwall boiler. The greater portion of the front wall is covered with refractory block. The other walls are covered with bare-faced blocks in the lower part and bare tubes in the upper part of the furnace.

The furnace water-walls, constituting the screen boiler, have a separate drum, the steam from this drum mixing with the steam from the boiler proper before entering the convection superheater.

Flames passing up into the boiler as far as the superheater indicated somewhat delayed combustion. An attempt was made to take gas analyses at points in the furnace in order to estimate the amount of this delayed combustion. Results obtained were extremely erratic, carbon monoxide in the middle of the furnace ranging anywhere from zero to 10 per cent. An analysis for hydrogen was not made. After numerous trials this gas sampling was given up as being too difficult for practical accomplishment because the time required to obtain representative samples would be longer than the boiler could be held at the given condition. This matter of delayed combustion was found to be characteristic of stoker-fired furnaces. No such extreme condition was evident in the furnaces fired by fuel in suspension.

Due to the difficulty encountered in the testing, tests were run only at high, medium, and low loads, the actual time of the tests being but a few minutes although sufficient time was allowed between tests to obtain fairly steady conditions.

The second boiler tested, although fired by pulverized coal, was similar to the first in that the bottom screen and side-wall tubes constituted an entirely separate boiler as may be seen by reference to Fig. 2. In this type of firing, the fuel is conveyed from the burners by a small amount of primary air. Secondary air is drawn in through ports in the side wall adjacent to the burners, located in this case, among the side-wall tubes. In order to assure a sufficiently long flame at the higher combustion rates, a small amount of preheated forced-draft air is introduced under

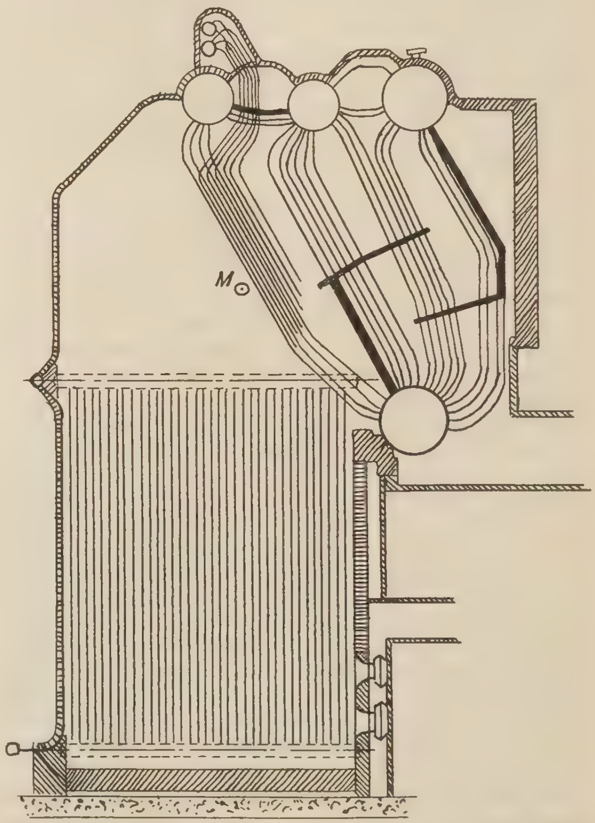

Fig. 4 All Water Cooled Combination Gas- and Oil-Fired Boiler

pressure at the burners. Because of this, the length of the flame is a function of both the primary and forced-draft air pressures.

Due to the operating and control conditions on this boiler it was feasible to make more extensive tests than were made on the first boiler. On this account each test was repeated for three excess-air values in order to determine the resulting effect.

Included in the first group of tests are results for three loads at three flame lengths and three excess-air values each, thus making a total of 27 tests. These tests were run half an hour apart, two readings being taken for each test. Gas analyses were also taken in the furnace during these tests. Although no carbon monoxide could be found, the gas-analysis results were so varied

that they were discarded and reliance was again placed on boiler-outlet gas-analysis values.

It should be noted that the terms "short flame," "normal flame," and "long flame" are only relative and do not have exactly the same magnitude for different tests.

As some of the results obtained in this first group of tests did not agree, it was thought that perhaps the duration of the tests was too short. It was then decided to run a second series of tests during which readings were taken every 10 or 15 minutes for a test period of 3 hours.

Comparison of the data obtained from this series of tests with the test data from boiler No. 1, showed that, in spite of the additional length of the tests, the results were not much better, and therefore the data have been omitted. The reason for this is probably the fact that the increased accuracy of the readings due to the longer time of the tests, is more than balanced by changed furnace conditions. Hence no more long-time tests were attempted on this boiler.

During the previous tests the side-wall tubes were covered with a certain amount of slag, the 2-in. space between the tubes and

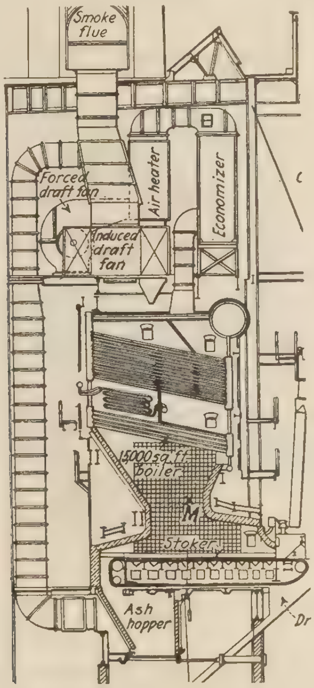

Fig. 5 Boiler Cooled on Two Sides of Furnace and Fired by Chain-Grate Stoker

the wall being completely filled with slag. In order to ascertain the effect of this slag on the screen-boiler evaporation, the boiler was taken off the line, the slag removed, and the tubes wire-brushed. Tests 2–8*C*, 2–14*C*, and 2–23*C*, were made under these new conditions.

The third boiler tested, Fig. 3, is exactly similar to boiler No. 2 except that there are no side-wall tubes and the superheater and baffling, with which we are not concerned, are slightly different. Comments applying to the boiler No. 2 apply also to boiler No. 3. Fewer tests were run on this boiler. From among the first group of tests, 3–8, 3–6, and 3–13 are given for three steam ratings.

As was done in the case of boiler No. 2, a second series of tests was run using longer test periods. The results were susceptible to the same conclusion as was arrived at in the case of the boiler No. 2, and are likewise omitted.

The fourth boiler tested was the oil- and gas-fired unit shown in Fig. 4. Oil was the fuel ordinarily used, natural gas being used as a stand-by fuel. The gases within the furnace were almost transparent. Due to operating conditions, a high load could not be carried on this boiler. Tests were made at three loads with two excess-air values each, for both oil- and gas-firing, except that test 4–12, which would have been with a high per cent air, was omitted. Duration of tests was approximately 1 hour.

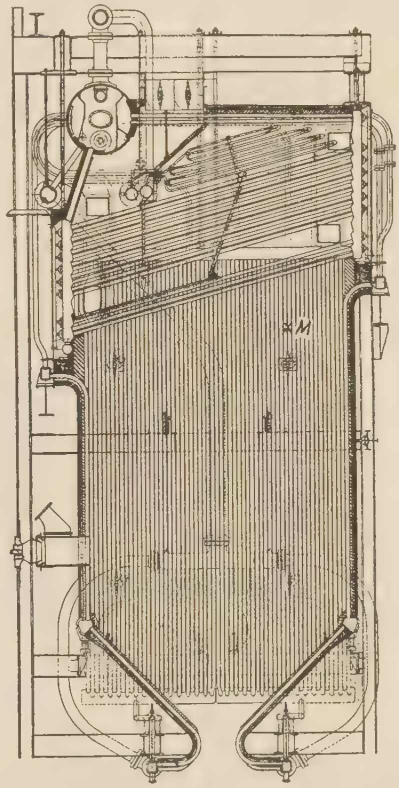

Fig. 6 All-Water-Cooled Pulverized-Coal Unit

The fifth boiler tested, shown in Fig. 5, was fired by a chain-grate stoker having two overhung combustion arches. The tests on this boiler were not as satisfactory as on the other boilers. Delayed combustion was again evident. Combustion was uneven on the grate and excess air was high. Instead of measuring the furnace-exit temperature just under the boiler tubes it was necessary to measure this temperature in the middle of the furnace as no other opening was available or could be made. It was not possible to vary the excess air, hence only three short tests were made at three boiler evaporations. These tests should not be considered as reliable as the other tests.

The sixth boiler tested, shown in Fig. 6, was an almost completely water-cooled furnace, burning pulverized coal and fired by two turbulent burners in the front wall. Due to temporary difficulty with the induced-draft fan, a high load could not be carried on the boiler. Similarly, a high excess-air medium load could not be carried. Hence, but five tests were run; three at different excess-air values at low load and two at different excess-air values at medium load. Duration of tests was from 1 to 2 hours.

The seventh boiler tested was the refractory-furnace gas- and oil-fired boiler shown in Fig. 7. Tests were made at three loads, when burning natural gas, operating with three excess-air values

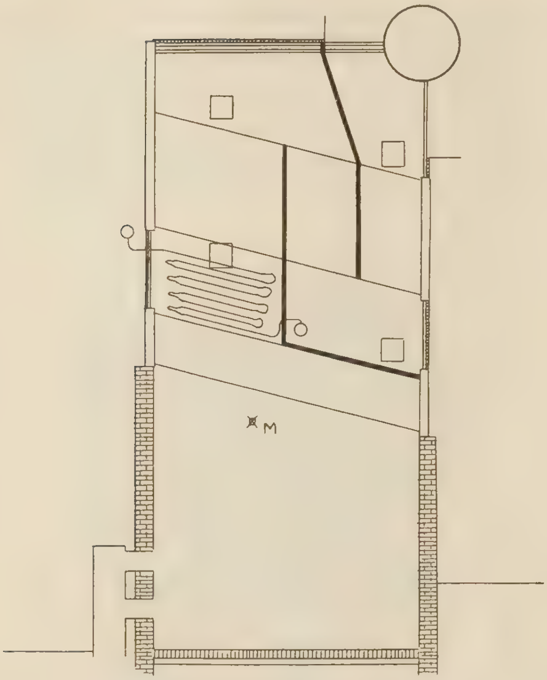

Fig. 7 Combination Gas- and Oil-Fired Boiler With All-Refractory Furnace

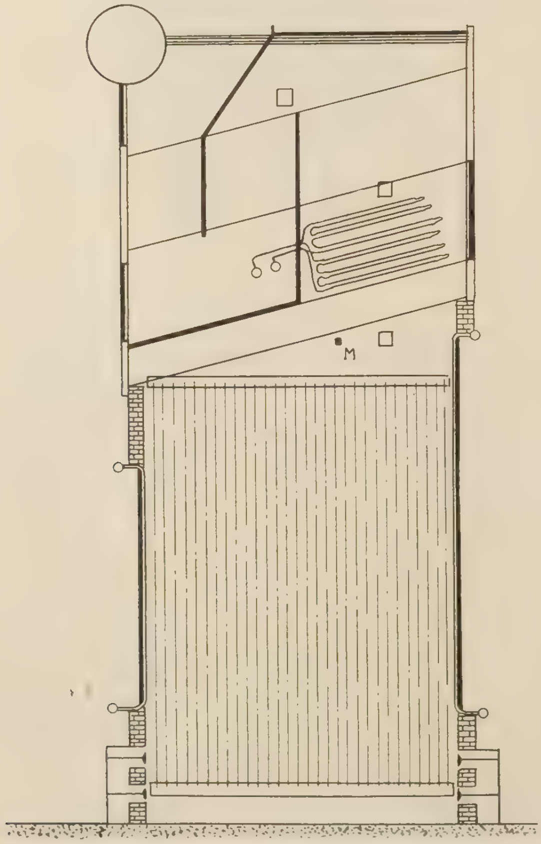

Fig. 8 Gas-Fired Boiler With Furnace Cooled on Four Sides

at each load except for the higher load where one test was omitted. Duration of tests was approximately half an hour with considerable time between tests.

The tests using oil (7–10, 7–11, and 7–12) were made at three boiler loads and were of limited duration because of the desire not to use oil, inasmuch as it was only an emergency or stand-by fuel. The furnace gases were transparent at all ratings.

The eighth boiler tested was the water-cooled natural-gas-fired boiler shown in Fig. 8. There was no provision for oil firing. Due to trouble in the turbine room, it was difficult to put high evaporative loads on this boiler. Tests 8–1 to 8–5 were run with low and medium loads and various air values. Tests 8–10, 8–11 and 8–12 were run at a medium boiler evaporation with different feedwater temperatures. Test 8–13 represents the one high-load test that was run. The furnace gases were perfectly transparent at all ratings. Each test required about 1 hour.

The ninth boiler was the underfeed-stoker-fired boiler with bare-tube waterwalls shown in Fig. 9. Tests 9–6 and 9–7 show

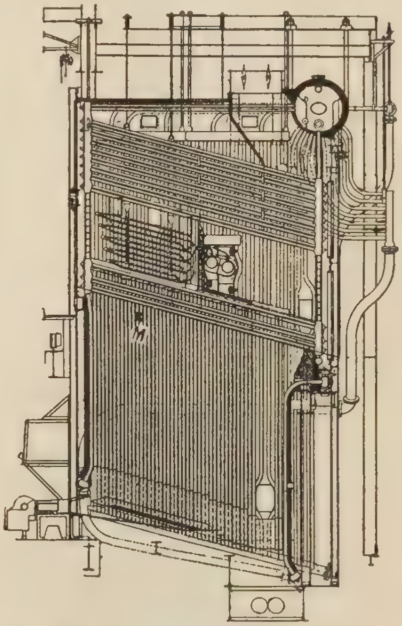

Fig. 9 Underfeed-Stoker-Fired Unit With All Water-Cooled Furnace

the effect of varying the feedwater temperature while holding the boiler at a constant output. Tests 9–6 and 9–7 were made on the boiler just before it was taken off the line for repairs. The tubes were dirty and the grates burned. Tests 9–14 to 9–19 were made after the grates had been reconditioned, the boiler having been in operation two days in order to obtain representative conditions. These tests were made at three evaporative ratings with two values of excess air each and each test lasted one-half hour.

The tenth boiler test was made on the large double-ended pulverized-coal-fired boiler shown in Fig. 10. This was the only boiler furnace in which any attempt was made to keep the side waterwall tubes clean during normal operation. The tubes were dusted twice daily by means of a compressed-air lance through four openings in each side wall, thus tending to retard slag formation on a portion of the waterwall surface.

Tests 10–12, 10–13, and 10–14 were run at three boiler loads with a low feedwater temperature. The feedwater temperature was increased and tests 10–15 to 10–20 followed at three loads with various excess-air values at each load. Each of these tests

were of 15-minutes duration. The data of a series of longer tests lasting perhaps three hours were discarded as being inconsistent, undoubtedly because boiler conditions change from day to day.

Discussion of the Tests

Several important points may be noted by examining the data obtained. Unless all of the tests are run in one day it is difficult to reconcile the data taken on two different days because of varying furnace conditions which are not a function of the operation of the boiler control instruments. In regard to future tests, it is believed that the influence of excess air has been amply demonstrated. Also, except for extremely accurate work, the importance of flame length in regard to furnace-exit temperature appears to be insufficient to warrant extended tests to show its effect.

As the object of these tests was to determine actual results which might be checked against theoretical calculations and existing formulas, it was advisable to determine values of the ratio of heat absorbed by radiation to the heat input to the furnace. These calculations are included in Table 3. The method of making the calculation is outlined in Table 4.

In calculations it is necessary to distinguish between the amount of excess air in the furnace and the amount at the boiler outlet. Using data obtained from a considerable number of tests, an average value of 7 per cent air leakage between the top of the first pass and the boiler outlet has been assumed. For calculation purposes the percentage of air at the top of the first pass has been assumed to be the same as that in the furnace. The percentages of air used as given in items 20 or 21 in Table 3, were obtained from calculated CO_2 curves for the fuels in question.

In order to determine the heat in the gases at the boiler outlet and at the furnace exit it was necessary to plot heat-capacity curves for the different fuels burned. A sample curve is given in Fig. 12. These curves are based on the values for the specific heat of gases recommended by Goodenough (58). Eastman's specific heats (57) are perhaps to be preferred and would result in an increase of about 2 per cent in the sensible heats. This difference in sensible-heat values affects the μ-values less than 2 per cent.

The initial temperature of the air, as well as the temperature at which the fuel is assumed to enter the furnace, has been taken as 80 F. The calculations are made as if the fuel entered the furnace at this temperature, the heat derived from combustion plus any preheat in the air being used to raise the products of combustion from 80 F to the actual furnace temperature. The sensible heat in the gases at any temperature therefore contains, not only any heat added by combustion, but also the heat in the

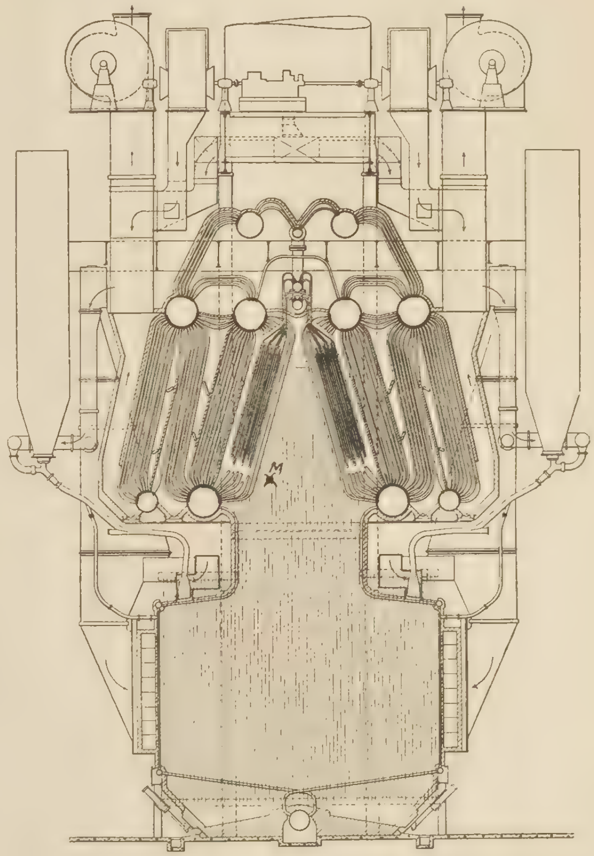

Fig. 10 Large Double-Ended Pulverized-Fuel Unit With Waterwalls on All Sides of Furnace

gases at 80 F, i.e., the heat-capacity curves are based upon 0 F abs.

The heat in the preheated air and the latent-heat losses have been calculated by conventional methods. The moisture is from (*a*) moisture in the air, (*b*) moisture in the coal, and (*c*) moisture resulting from the combustion of hydrogen and hydrocarbons in the fuel. The first does not occasion a latent-heat loss. The latter two require heat from the coal to vaporize the moisture.

It is to be noted that the energy as heat taken from the combustion heat to evaporate hydrogen and moisture into water vapor does not appear as British thermal units in the gases in the heat-capacity curves. Since the

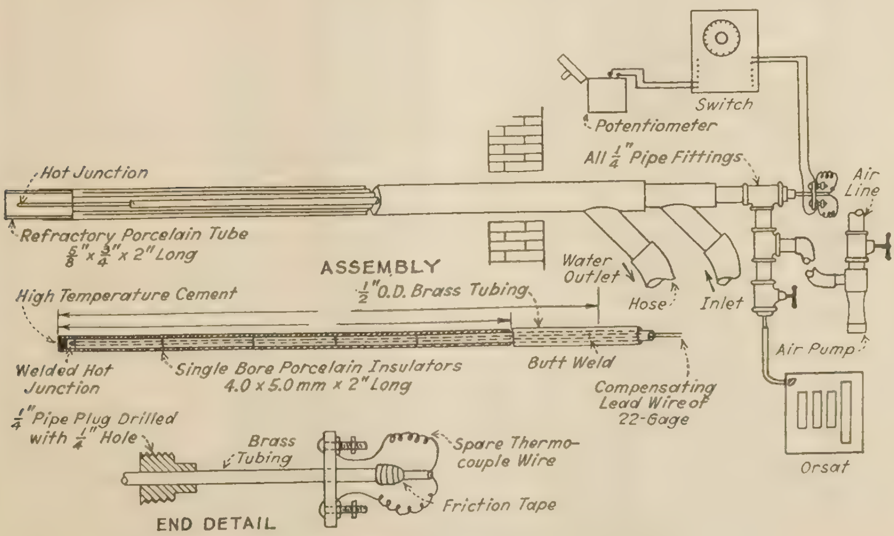

Fig. 11 High-Velocity Thermocouple With Orsat

TABLE 3 DATA OF TESTS INCLUDING FURNACE-EXIT

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
No. of boiler	Test number	Fuel	Flame conditions	Actual total boiler evap, 1000 lb per hr	Actual screen boiler evap, 1000 lb per hr	Furnace exit temperature, F	Boiler outlet temperature, F	Steam pressure, lb per sq in., abs	Feedwater temperature, F	Steam temperature, F	Forced-draft pressure, in. water	Primary air pressure, in. water	Room temp, F	Air temperature at burner, F	Gas analysis at first pass or outlet	Carbon dioxide, %	Oxygen, %	Carbon monoxide, %	Furnace air, %	Boiler outlet air calculated, %
1[a]	1-1	coal	..	300	124	2800	788	413	380	688	3.9	...	...	362	outlet	15.0	...	0	116	123
	1-2	coal	..	215	98	2575	688	412	372	683	2.9	...	...	325	outlet	12.6	...	0	139	146
	1-3	coal	..	115	55	2140	568	415	350	654	0.5	...	...	310	outlet	13.5	...	0	130	137
2[b]	2-1	coal	S[f]	150	58	2150	560	395	335	715	0.1	6.0	65	435	outlet	13.1	6.0	0	133	140
	2-2	coal	S	150	63	2150	555	393	335	707	0.1	4.0	66	428	outlet	12.8	6.2	0	136	143
	2-3	coal	S	150	65	2155	550	398	330	705	0.1	4.0	63		outlet	14.5	4.4	0	121	127
	2-4	coal	N[g]	150	60	1970	550	390	325	710	1.4	4.0	63	...	outlet	12.0	6.5	0	146	153
	2-5	coal	N	150	68	2080	550	388	325	710	0.4	6.0	69	...	outlet	12.8	6.1	0	136	143
	2-6	coal	N	150	64	2100	550	397	330	703	0.1	4.0	69	...	outlet	13.5	5.2	0	128	135
	2-7	coal	L[h]	150	62	2035	550	397	330	710	0.4	9.0	69	...	outlet	12.7	6.2	0	138	145
	2-8	coal	L	150	60	2005	550	394	330	710	0.1	8.0	69	423	outlet	12.7	6.3	0	138	145
	2-9	coal	L	150	62	2100	550	396	330	703	0.1	7.0	73	435	outlet	13.4	5.5	0	129	136
	2-10	coal	S	225	66	2390	610	393	340	750	0.1	5.0	74	475	outlet	13.8	5.2	0	126	133
	2-11	coal	S	228	72	2435	610	395	340	745	0.1	5.0	74	480	outlet	15.1	3.7	0	115	122
	2-12	coal	S	225	75	2460	600	405	330	738	0.1	5.0	74	460	outlet	14.7	4.3	0	118	125
	2-13	coal	N	225	84	2310	606	410	335	735	1.4	11.0	69	490	outlet	14.5	4.5	0	120	127
	2-14	coal	N	235	90	2325	590	385	325	723	1.4	9.0	67	424	outlet	15.2	3.5	0	115	122
	2-15	coal	N	230	93	2350	580	391	325	723	0.7	7.0	70	447	outlet	14.1	4.5	0	123	130
	2-16	coal	L	225	84	2175	570	380	335	732	2.6	13.0	72	418	outlet	12.0	6.8	0	146	153
	2-17	coal	L	225	88	2275	570	405	330	720	2.5	14.0	73	400	outlet	13.8	5.1	0	126	133
	2-18	coal	L	225	94	2245	560	391	325	720	2.0	12.0	73	410	outlet	15.3	3.5	0	114	121
	2-19	coal	S	270	75	2485	630	387	360	763	0.1	9.0	75	490	outlet	13.5	5.7	0	129	136
	2-20	coal	S	275	78	2525	620	390	355	767	0.2	9.0	76	495	outlet	13.6	5.4	0	128	125
	2-22	coal	N	280	82	2390	630	394	355	762	1.4	9.0	77	482	outlet	13.9	5.0	0	125	132
	2-23	coal	N	280	92	2425	615	392	350	752	1.4	9.0	77	442	outlet	14.6	4.3	0	119	126
	2-24	coal	N	280	104	2480	600	398	325	747	1.4	9.0	78	435	outlet	16.0	2.9	0	108	105
	2-25	coal	L	280	95	2350	615	402	345	750	3.0	11.0	78	448	outlet	13.7	5.3	0	127	134
	2-26	coal	L	280	101	2305	610	390	340	744	2.7	11.0	77	422	outlet	14.0	5.0	0	125	132
	2-27	coal	L	285	106	2360	600	398	330	737	2.8	11.0	78	418	outlet	15.7	3.1	0	110	117
	2-8*C*	coal	..	147	70.4	2000	528	...	...	...	0.1	8.8	88	...	...	...	...	...	...	...
	2-14*C*	coal	..	229	100.0	2265	576	...	...	...	1.0	8.5	86	...	...	...	...	...	...	...
	2-23*C*	coal	..	281	113.0	2375	630	...	...	...	1.4	10.0	82	...	...	...	...	...	...	...
3[b]	3-8	coal	N	150	53.7	2170	520	400	315	691	0.1	5.0	74	...	first	13.9	5.1	0	132	139
	3-6	coal	N	225	71.0	2320	570	407	330	687	0.8	11.0	72	...	first	14.7	4.1	0	125	132
	3-13	coal	N	285	70.5	2360	645	411	350	726	1.2	12.5	74	...	first	13.8	5.2	0	133	140
4[c]	4-1	oil	..	70	...	1600	575	440	274	632	...	...	105	500	outlet	12.0	6.3	0	133	140
	4-2	oil	..	73	...	1600	585	440	276	640	...	...	105	510	outlet	10.8	7.7	0	148	155
	4-3	oil	..	114	...	1880	630	445	270	657	...	...	97	535	outlet	14.1	3.3	0	112	119
	4-4	oil	..	122	...	1985	665	445	270	680	...	...	97	557	outlet	12.8	4.1	0	124	131
	4-5	oil	..	95	...	1610	595	440	270	618	...	...	97	514	outlet	13.8	4.0	0	115	122
	4-6	oil		94	...	1615	632	440	270	637	...	...	97	530	outlet	11.5	7.0	0	139	146
	4-7	gas	..	114	...	1910	638	442	274	670	...	...	100	545	outlet	9.7	4.0	0.2	114	121
	4-8	gas	..	115	...	1930	680	442	274	684	...	...	100	573	outlet	7.4	8.3	0.1	149	156
	4-9	gas	..	89	...	1785	598	445	274	650	...	...	100	531	outlet	8.7	5.7	0.1	127	134
	4-10	gas	..	87	...	1780	622	445	276	670	...	...	100	540	outlet	7.4	7.9	0.1	149	156
	4-11	gas	..	71	...	1730	575	440	276	650	...	...	100	513	outlet	8.8	5.5	0.1	125	132
5[d]	5-1	coal	..	49.5	...	1905	420	380	295	662	...	...	113	230	outlet	9.0	10.2	0	198	205
	5-2	coal	..	91	...	2325	554	380	315	717	...	...	110	275	outlet	9.8	9.5	0	182	189
	5-3	coal	..	128	...	2415	625	390	320	742	...	...	109	302	outlet	12.4	6.5	0.2	144	151
6[b]	6-1	coal	..	65	...	1790	557	456	217	577	2.17	0.75	67	405	outlet	12.4	...	...	145	152
	6-2	coal	..	63	...	1885	540	456	216	565	15.0	0.60	62	412	outlet	14.0	...	...	128	135
	6-3	coal	..	61	...	1845	526	456	216	556	10.0	0.13	62	416	outlet	15.8	...	...	113	120
	6-4	coal	..	99	...	1885	610	463	218	587	6.5	0.80	79	430	outlet	15.6	...	...	115	122
	6-5	coal	..	99	...	1910	575	463	221	559	4.1	0.75	79	403	outlet	17.2	...	...	103	110
7[c]	7-1	gas	..	60	...	2175	500	360	233	678	5.0	...	96	96	first	7.8	...	...	145	152
	7-2	gas	..	60	...	2205	490	359	230	664	5.3	...	96	96	first	9.1	...	...	125	132
	7-3	gas	..	60	...	2255	490	360	233	651	5.2	...	96	96	first	9.9	...	...	116	123
	7-4	gas	..	120	...	2400	600	363	234	725	4.0	...	102	102	first	8.6	...	...	132	139
	7-5	gas	..	121	...	2430	590	365	243	720	4.0	...	94	94	first	9.1	...	...	125	132
	7-6	gas	..	121	...	2490	580	364	242	715	4.4	...	94	94	first	9.9	...	...	116	123
	7-8	gas	..	176	...	2640	710	367	230	750	7.0	...	101	101	first	9.1	...	...	125	132
	7-9	gas	..	184	...	2700	710	367	238	750	6.5	...	101	101	first	9.6	...	...	118	125
	7-10	oil	..	68	...	2190	505	363	232	682	7.2	...	87	87	first	10.6	...	...	159	166
	7-11	oil	..	116	...	2410	615	366	233	712	6.8	...	87	87	first	11.5	...	...	146	153
	7-12	oil	..	160	...	2475	680	368	227	739	7.8	...	87	87	first	10.9	...	...	154	161
8[e]	8-1	gas	..	100	...	1970	495	395	284	637	...	...	67	67	first	10.7	1.5	...	109	116
	8-2	gas	..	102	...	1955	510	397	285	652	...	...	67	67	first	9.7	3.4	...	120	127
	8-3	gas	..	101	...	2010	625	399	286	669	...	...	68	68	first	8.7	5.2	...	134	141
	8-4	gas	..	201	...	2095	630	407	289	710	...	...	69	69	first	10.6	1.9	...	110	117
	8-5	gas	..	198		1920	646	407	289	740	...	...	69	69	first	9.4	4.2	...	123	130
	8-10	gas	..	204	...	2060	630	407	302	708	...	...	69	69	first	10.6	1.9	...	110	117
	8-11	gas	..	203	...	2345	640	405	246	740	...	...	69	69	first	10.6	1.8	...	110	117
	8-12	gas	..	206	...	2335	640	404	218	750	...	...	69	69	first	10.6	1.8	...	110	117
	8-13	gas	..	297	...	2315	714	407	309	740	...	...	69	69	first	10.6	2.3	...	110	117
9[a]	9-6	coal	..	225		2330	557	383	336	698	3.9	...	...	404	first	14.0	5.4	0	134	141
	9-7	coal	..	225	...	2310	551	383	221	721	4.5	...	...	403	first	15.0	3.1	0	125	132
	9-14	coal	..	127		2005	472	377	332	747	2.0	...	...	336	first	11.5	7.5	0	163	170
	9-15	coal	..	125	...	1920	462	375	331	713	1.9	...	...	328	first	13.0	6.0	0	144	151
	9-16	coal		224	...	2200	554	379	332	774	3.2	...	...	371	first	11.6	8.0	0	161	168
	9-17	coal	..	230	...	2200	538	379	341	742	3.3	...	...	375	first	11.5	8.3	0	163	170
	9-18	coal	..	275	...	2460	578	382	332	763	3.7	...	...	410	first	13.0	6.2	0	144	151
	9-19	coal		275		2495	575	384	334	775	3.8	...	...	407	first	12.7	5.3	0	147	154
10[b]	10-12	coal	..	800		2345	685	407	206	771	...	15.2	78	495	outlet	16.2	...	...	109	116
	10-13	coal	..	615	...	2215	630	407	203	740	...	12.7	78	465	outlet	14.7	...	...	121	128
	10-14	coal	..	415	...	2040	570	407	207	716	...	11.1	78	440	outlet	13.1	...	...	137	144
	10-15	coal	..	415		1980	570	407	287	714	...	10.7	78	450	outlet	12.7	...	...	141	148
	10-16	coal	..	415		1950	585	408	289	723	...	11.5	78	450	outlet	10.6	...	...	170	177
	10-17	coal	..	590	...	2305	640	407	291	770	...	13.7	78	475	outlet	13.6	...	...	132	139
	10-18	coal	..	600	...	2095	645	407	291	737	...	14.5	78	480	outlet	12.6	...	...	142	149
	10-19	coal	..	620		2135	640	407	293	733	...	14.3	78	480	outlet	14.2	...	...	126	133
	10-20	coal	..	730	...	2270	690	407	292	743	...	17.0	78	500	outlet	15.3	...	...	116	123

[a] Equipped with underfeed stoker. [b] Burns pulverized coal. [c] Equipped to fire oil or gas. [d] Equipped with chain-grate stoker. [e] Burns natural gas.

TEMPERATURES AND CALCULATED μ-VALUES

22 Sensible heat in furnace gases at boiler outlet, 1000 Btu per lb wf[i]	23 Sensible heat in gases at 80 F, 1000 Btu per lb wf	24 Heat in preheated air, 1000 Btu per lb wf	25 (High heat fuel + air heat), 1000 Btu per lb wf	26 (High heat fuel + air heat — latent heat), 1000 Btu per lb wf	27 (High heat fuel + air heat + gas heat at 80 F — latent heat), 1000 Btu per lb wf	28 Combustible + boiler radiation loss, %	29 Heat loss in dry gases at boiler outlet, 1000 Btu per lb wf	30 Summation of losses	31 Boiler efficiency based on furnace input	32 Heat added by boiler to total steam, 10^6 Btu per hr	33 Heat input to furnace, 10^6 Btu per hr	34 Heat in gases at furnace temp, 1000 Btu per lb wf	35 Heat to furnace surface, 1000 Btu per lb wf	36 Gas temp μ based on [(high heat fuel) + (air heat)]	37 Gas temp μ based on [(low heat fuel) + (air heat)]	38 Furnace heat input per unit volume, 1000 Btu per cu ft
4.35	1.70	0.83	14.68	14.18	15.90	5.0	2.65	0.265	0.735	301	410	11.7	4.20	0.29	0.30	29.3
4.65	2.05	0.83	14.68	14.18	16.25	4.2	2.60	0.254	0.746	216	290	12.5	3.75	0.25	0.26	20.7
3.90	2.00	0.83	14.61	14.11	16.10	3.5	1.90	0.194	0.806	117	145	9.9	6.20	0.42	0.44	10.3
3.80	1.90	0.25	13.70	13.19	15.10	2.5	1.90	0.201	0.799	160	200	9.7	5.40	0.39	0.41	8.9
3.85	2.00	0.25	13.70	13.19	15.20	2.5	1.85	0.197	0.802	160	200	9.8	5.40	0.39	0.41	8.9
3.40	1.80	0.25	13.70	13.19	15.00	2.5	1.60	0.179	0.821	160	195	8.8	6.20	0.45	0.47	8.7
4.00	2.10	0.25	13.70	13.19	15.30	2.5	1.90	0.200	0.800	161	201	9.7	5.60	0.41	0.42	8.9
3.85	2.00	0.25	13.70	13.19	15.20	2.5	1.85	0.197	0.803	161	200	9.6	5.60	0.41	0.42	8.9
3.60	1.85	0.25	13.70	13.19	15.25	2.5	1.75	0.190	0.810	160	198	9.0	6.25	0.45	0.47	8.8
3.85	2.00	0.25	13.70	13.19	15.20	2.5	1.85	0.197	0.803	160	199	9.5	5.70	0.42	0.43	8.9
3.85	2.00	0.25	13.70	13.19	15.20	2.5	1.85	0.197	0.803	160	199	9.4	5.80	0.42	0.44	8.9
3.65	1.85	0.25	13.70	13.19	15.05	2.5	0.80	0.193	0.807	160	198	9.2	5.85	0.43	0.44	8.8
3.80	1.85	0.25	13.70	13.19	15.05	3.2	1.95	0.212	0.788	242	307	10.2	4.85	0.35	0.37	13.7
3.50	1.70	0.25	13.70	13.19	14.90	3.2	1.80	0.200	0.800	238	298	9.6	5.30	0.39	0.40	13.3
3.50	1.75	0.25	13.70	13.19	14.95	3.2	1.75	0.196	0.804	244	304	9.8	5.15	0.38	0.39	13.6
3.60	1.75	0.25	13.70	13.19	14.95	3.2	1.85	0.204	0.796	242	304	9.4	5.55	0.41	0.42	13.6
3.45	1.70	0.25	13.70	13.19	14.90	3.2	1.75	0.197	0.803	254	316	9.2	5.70	0.42	0.43	14.1
3.60	1.80	0.25	13.70	13.19	15.00	3.2	1.80	0.201	0.799	249	312	9.8	5.20	0.38	0.39	13.9
4.15	2.10	0.25	13.70	13.19	15.30	3.2	2.05	0.219	0.781	242	310	10.6	4.70	0.34	0.36	13.8
3.65	1.85	0.25	13.70	13.19	15.05	3.2	1.80	0.201	0.799	241	302	9.7	5.35	0.39	0.40	13.5
3.30	1.70	0.25	13.70	13.19	14.90	3.2	1.60	0.186	0.814	242	298	8.8	6.10	0.45	0.46	13.3
3.90	1.85	0.25	13.70	13.19	15.05	4.0	2.05	0.226	0.774	288	372	10.7	4.35	0.32	0.33	16.6
3.85	1.85	0.25	13.70	13.19	15.05	4.0	2.00	0.223	0.777	295	380	10.8	4.25	0.31	0.32	17.0
3.85	1.80	0.25	13.70	13.19	15.00	4.0	2.05	0.226	0.774	300	388	10.2	4.80	0.35	0.36	17.3
3.65	1.75	0.25	13.70	13.19	14.95	4.0	1.90	0.215	0.785	300	382	9.8	5.15	0.38	0.39	17.1
3.30	1.60	0.25	13.70	13.19	14.80	4.0	1.70	0.201	0.799	303	379	9.3	5.50	0.40	0.42	16.9
3.85	1.85	0.25	13.70	13.19	15.00	4.0	2.00	0.222	0.778	301	387	10.0	5.00	0.36	0.38	17.3
3.80	1.80	0.25	13.70	13.19	15.00	4.0	2.00	0.222	0.778	302	388	9.8	5.20	0.38	0.39	17.3
3.30	1.70	0.25	13.70	13.19	14.90	4.0	1.60	0.193	0.807	309	383	9.0	5.90	0.43	0.45	17.1
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3.60	1.90	0.25	13.70	13.19	15.10	2.5	1.70	0.186	0.814	161	198	9.7	5.40	0.39	0.41	8.9
3.60	1.80	0.25	13.70	13.19	15.00	3.2	1.80	0.201	0.799	236	296	9.8	5.20	0.38	0.39	13.2
4.10	1.90	0.25	13.70	13.19	15.10	4.0	2.20	0.237	0.763	301	394	10.6	4.50	0.33	0.34	17.6
5.10	2.60	1.73	20.15	19.30	21.85	3.0	2.50	0.196	0.804	75.5	94	10.1	11.70	0.58	0.61	8.5
5.60	2.00	2.04	20.46	19.61	21.60	3.0	2.70	0.235	0.765	79.0	103	11.1	10.50	0.51	0.53	9.3
4.65	2.25	1.62	20.04	19.19	21.45	3.0	2.40	0.192	0.808	125	155	10.0	11.45	0.57	0.60	14.1
5.15	2.40	1.90	20.32	19.47	21.85	3.0	2.75	0.208	0.792	133	168	11.4	10.45	0.51	0.54	15.3
4.50	2.30	1.25	19.67	18.82	21.10	3.0	2.20	0.185	0.815	102	125	8.9	12.20	0.62	0.65	11.4
5.55	2.70	2.00	20.42	19.57	22.25	3.0	2.85	0.211	0.789	102	129	10.6	11.65	0.57	0.60	11.7
5.90	2.80	2.00	24.50	22.32	25.00	3.0	3.10	0.246	0.754	125	166	12.9	12.10	0.50	0.54	15.1
7.70	3.50	2.76	25.16	22.98	26.50	3.0	4.20	0.284	0.716	127	177	16.3	10.20	0.41	0.45	16.1
6.30	3.20	2.15	24.65	22.47	25.60	3.0	3.10	0.244	0.756	96.8	128	13.1	12.50	0.51	0.55	11.6
7.20	3.50	2.60	25.10	22.92	26.40	3.0	3.70	0.264	0.736	95.5	129	15.2	11.20	0.45	0.49	11.7
6.00	3.10	2.04	24.54	22.36	25.40	3.0	2.90	0.238	0.762	77.2	101	12.8	12.60	0.52	0.56	9.2
2.90	1.70	0.46	8.71	8.21	9.90	4.0	1.20	0.235	0.765	53.4	70	8.2	1.70	0.20	0.21	12.9
3.20	1.50	0.52	8.77	8.27	9.75	4.5	1.70	0.295	0.705	99.0	140	9.0	0.75	0.08	0.09	25.8
2.80	1.30	0.48	8.75	8.23	9.50	5.0	1.50	0.280	0.720	140	194	7.7	1.80	0.21	0.22	35.8
2.75	1.40	0.77	10.82	10.34	11.70	4.0	1.25	0.200	0.800	71.7	89.6	6.2	5.50	0.51	0.53	10.3
2.40	1.35	0.65	10.70	10.22	11.55	4.0	1.05	0.183	0.817	69.0	84.5	5.8	5.70	0.53	0.56	9.7
2.15	1.15	0.60	10.65	10.17	11.30	4.0	1.00	0.179	0.821	66.5	81	5.2	6.10	0.57	0.60	9.3
2.35	1.20	0.63	10.68	10.20	11.40	4.0	1.15	0.202	0.798	109	136	5.3	6.10	0.57	0.60	15.6
2.10	1.05	0.56	10.61	10.13	11.25	5.0	1.05	0.194	0.806	107	132	5.0	6.25	0.59	0.62	15.1
5.70	3.10	0	20.60	18.60	22.40	3.0	2.60	0.245	0.755	69.2	91.7	16.1	5.60	0.27	0.30	16.5
5.00	2.80	0	20.60	18.60	22.10	3.0	2.20	0.226	0.774	68.8	89.8	14.4	7.00	0.34	0.38	16.0
4.90	2.60	0	20.60	18.60	21.90	3.0	2.30	0.231	0.769	68.2	88.8	13.9	7.30	0.35	0.39	16.0
6.00	3.00	0	20.60	18.60	22.30	2.5	3.00	0.259	0.741	141	190	16.3	5.30	0.26	0.29	34.2
5.60	2.90	0	20.60	18.60	22.20	2.5	2.70	0.245	0.755	141	187	15.7	5.80	0.28	0.31	33.7
5.30	2.60	0	20.60	18.60	21.90	2.5	2.70	0.245	0.755	141	187	15.0	6.20	0.30	0.33	33.7
6.40	2.90	0	20.60	18.60	22.20	2.0	3.50	0.278	0.722	210	291	17.0	4.50	0.22	0.24	52.4
6.10	2.60	0	20.60	18.60	21.90	2.0	3.50	0.278	0.722	218	302	16.3	4.90	0.24	0.26	54.3
5.50	3.00	0	19.00	18.15	21.15	3.0	2.50	0.205	0.795	78.5	98.8	15.5	5.65	0.30	0.31	7.8
5.55	2.85	0	19.00	18.15	21.15	2.5	2.70	0.210	0.790	135	171	15.7	5.45	0.29	0.30	30.8
6.40	2.90	0	19.00	18.15	21.20	2.0	3.50	0.247	0.753	190	252	17.1	4.10	0.22	0.23	45.4
4.20	2.50	0	20.00	18.11	20.60	2.5	1.70	0.204	0.796	117	147	11.1	9.50	0.48	0.52	11.1
4.20	2.60	0	20.00	18.11	20.70	2.5	2.10	0.224	0.776	120	154	12.4	8.60	0.43	0.47	11.7
5.90	2.80	0	20.00	18.11	20.90	2.5	3.10	0.274	0.726	120	165	13.7	7.20	0.36	0.40	12.5
5.00	2.50	0	20.00	18.11	20.60	2.0	2.50	0.239	0.761	223	293	11.9	8.70	0.43	0.48	22.2
5.60	2.70	0	20.00	18.11	20.80	2.0	2.90	0.259	0.741	156	210	12.3	8.50	0.43	0.47	15.9
5.20	2.40	0	20.00	18.11	20.50	2.0	2.80	0.254	0.746	224	300	11.7	8.80	0.44	0.49	23.7
5.00	2.40	0	20.00	18.11	20.50	2.0	2.60	0.244	0.756	162	214	13.2	7.50	0.36	0.40	16.2
5.00	2.40	0	20.00	18.11	20.50	2.0	2.60	0.244	0.756	248	328	13.2	7.50	0.36	0.40	24.9
5.30	2.40	0	20.00	18.11	20.50	2.0	2.90	0.259	0.741	328	442	13.1	7.40	0.37	0.41	33.7
3.90	2.00	1.08	15.23	14.79	16.80	4.2	1.90	0.195	0.805	238	296	11.0	5.80	0.38	0.39	32.7
3.65	1.90	0.97	15.12	14.68	16.60	4.2	1.75	0.187	0.813	267	328	10.2	6.40	0.42	0.44	36.3
4.30	2.40	1.05	15.20	14.76	17.20	3.5	1.90	0.189	0.811	138	170	11.0	6.20	0.41	0.42	18.8
3.80	2.15	0.87	15.02	14.58	16.80	3.5	1.65	0.174	0.826	134	162	9.8	7.00	0.46	0.48	17.9
4.60	2.40	1.19	15.34	14.90	17.30	4.2	2.20	0.214	0.786	247	314	12.0	5.30	0.34	0.36	34.7
4.60	2.40	1.15	15.30	14.86	17.30	4.2	2.20	0.215	0.785	247	314	12.1	5.20	0.34	0.35	34.7
4.30	2.15	1.15	15.30	14.86	17.05	5.0	2.15	0.219	0.781	301	385	12.3	4.75	0.31	0.32	42.6
4.40	2.20	1.19	15.34	14.90	17.15	5.0	2.20	0.222	0.778	302	388	12.7	4.45	0.29	0.30	42.9
3.70	1.70	1.14	15.69	15.28	17.10	5.0	2.00	0.204	0.796	980	1230	9.3	7.80	0.50	0.51	36.8
4.00	1.80	1.10	15.65	15.24	17.05	4.0	2.20	0.207	0.793	747	942	9.7	7.35	0.47	0.48	28.2
4.10	2.00	1.14	15.69	15.28	17.30	3.0	2.10	0.190	0.800	496	620	10.0	7.30	0.46	0.48	18.6
4.20	2.00	1.32	15.87	15.45	17.50	3.0	2.20	0.194	0.806	463	574	10.1	7.40	0.47	0.48	17.2
4.80	2.30	1.01	16.16	15.75	18.05	3.0	1.50	0.148	0.832	464	544	11.7	6.35	0.39	0.40	16.3
4.20	1.90	1.24	15.79	15.38	17.30	4.0	2.30	0.212	0.788	674	855	10.8	6.50	0.41	0.42	25.6
4.50	2.00	1.43	15.98	15.57	17.60	4.0	2.50	0.222	0.778	674	866	10.7	6.90	0.43	0.44	26.0
4.10	1.80	1.17	15.72	15.31	17.10	4.0	2.30	0.212	0.788	693	887	9.7	7.40	0.47	0.48	26.4
4.00	1.70	1.28	15.83	15.42	17.10	5.0	2.30	0.221	0.779	877	1125	9.4	7.70	0.49	0.50	33.7

[f] Short-flame conditions. [g] Normal-flame conditions. [h] Long-flame conditions. [i] Wet fuel.

TABLE 4 INDICATED CALCULATIONS FOR TABLE 3

Items 1 to 19	data
Item 20	calculated from item 17, using fuel analysis
Item 21	assumed equal to item 20 plus 7 per cent
Item 22	heat capacity curves using items 8 and 21
Item 23	heat capacity curves using item 20
Item 24	(fraction of air heated) × [(heat in air at burner temp) — (heat in air at 80 F)]
Item 25	high heat value of fuel + item 24
Item 26	item 25 — latent heat
Item 27	item 26 + item 23
Item 28	assumed
Item 29	item 22 — item 23
Item 30	$\frac{\text{latent-heat loss in per cent} + \text{item 28}}{100} + \frac{\text{item 29}}{\text{item 25}}$
Item 31	1.00 — item 30
Item 32	item 5 × [(heat of superheated steam — heat of feed-water)]
Item 33	item 32/item 31
Item 34	heat-capacity curves using items 7 and 20
Item 35	item 27 — item 34
Item 36	item 35/item 25
Item 37	item 35/item 26
Item 38	item 33/furnace volume

TABLE 5 ASSUMED RADIATION LOSSES

Boiler surface, sq ft	Per cent loss
2,500 to 7,500	3.00
7,500 to 12,500	2.00
12,500 to 20,000	1.50
20,000 to 50,000	1.25
Over 50,000	1.00

heat is lost in so far as any heat transfer is concerned, unless the dewpoint of the gases is reached, it must be subtracted from the high heat value of the fuel.

It was impossible to conduct a weighed-fuel and water test on the boilers, thus necessitating the use of the heat-balance method (38) for the calculation of boiler efficiency. Results should be sufficiently close for the purposes of this report. The carbon, hydrogen, hydrocarbon, and overall boiler radiation loss have been combined and estimated in item 28 of Table 3.

The radiation loss through the furnace walls in per cent of the furnace heat input was taken as in Table 5.

The other losses were, judging by some experimental data (30), estimated as about 1 per cent for oil- and gas-fired boilers. For coal firing, these losses were estimated at values ranging from 2 per cent at normal ratings to 6 per cent at the higher ratings, these values being based on available data.

The calculations made in Table 3 to obtain the heat input to the furnace are evident. Using the heat-capacity curves in order to determine the heat in the gases at the furnace exit, the difference between this value and the heat of combustion plus the heat in the gases at 80 F plus the heat in the preheated air, will be the heat radiated or convected to the furnace surfaces. Complete combustion is assumed for this calculation. The ratio of the heat radiated to the heat input is readily calculated, and is termed the gas-temperature μ-value.

It will be seen that the heat in the preheated air used for combustion has been assumed to act exactly as if it were part of the calorific value of the fuel. This assumption is open to question in view of the possibility of chemiluminescence but no data exist to form a more exact conclusion. Any error from this source cannot be large.

There are two possible methods of computing the gas-temperature μ-value. In item 36 it is given as a ratio of the heat given up from the furnace gases to the calorific heat of the fuel plus sensible heat in air supplied from combustion.

Since the energy as heat required to evaporate the moisture resulting from hydrogen and moisture in the fuel is not available for radiation or to increase the sensible heat of the gases, a second determination of μ as a ratio of the heat given up from the furnace gases to the low heat value of the fuel plus any air preheat has been given as item 37.

In the cases where the boilers tested have separate waterwall circulation and drums, it is possible to estimate the heat absorbed in the furnace by use of a proportion between the total exposed effective radiant heating surface in the screen boiler and that part of the waterwall boiler on separate circulation.

This has not been done in this report since it involves an estimation of the relative heat-absorbing abilities of various kinds of waterwall surfaces which, in the present state of our knowledge of this subject, is to some extent dependent on individual opinion.

It should be noted that in calculating the gas-temperature μ-values, it was necessary to determine the boiler efficiency by the heat-balance method. This involves some element of personal judgment and it is realized that others would perhaps obtain different values for the boiler efficiency, thus getting slightly different μ-values. However, the differences should not be large and it has been deemed advisable to calculate and give these μ-values so as to present the data more effectively.

The data which are included will also serve, it is believed, as a criterion for checking results obtained by the various methods of calculation advanced by different investigators.

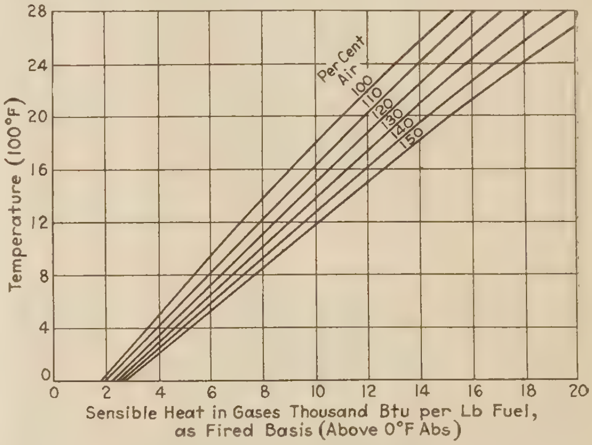

FIG. 12 TYPICAL HEAT-CAPACITY CURVES FOR NATURAL GAS
(Goodenough specific heats used.)

With this in view the values of μ as computed from gas-temperature data have been included in Table 3, as stated previously. It is felt that the results included in a report by the committee should not be carried beyond this point because major assumptions would be involved about which there is no general agreement.

However, an investigation has been conducted in which the results of each of several well-known methods of arriving at μ-values are compared to the gas-temperature values of μ included in this report. This information is presented in the two papers, immediately preceding this paper, RP-57-2 and RP-57-3.

A study of the method of taking the experimental data included in this report will reveal the probable magnitude of any errors involved. The readings of the high-velocity thermocouples, by far the most important determination, are believed to indicate the true gas temperature within 50 F.

Furthermore, inspection of the data (of which a considerable amount is not given in this paper) will show that even under identical conditions of boiler-operating control, the furnace temperature may vary as much as 100 F from day to day. Theoretically, it would be possible to determine the cause of this variation which is probably due to such factors as irregular coal feeding, shifting combustion conditions, and different amounts of slag on the walls. Practically, the causes are indeterminate, hence, it would seem that a method yielding the greatest accuracy desirable should give results, in regard to the furnace temperature, of about ±50 F.

It is appreciated that many faults may be found with the

results. However, it is believed that a contribution has been made which will help in advancing the subject but which will require much more work, both practical and theoretical, before it can be said that it is on a thoroughly sound basis.

In the literature, the increasing tendency to measure furnace-exit-gas temperatures as a part of the complete boiler test is noted with gratification (30, 35, 50), although it is regrettable that high-velocity thermocouples are not always used for this determination.

It is impossible to mention individually all those who have cooperated in this investigation. Grateful acknowledgment is particularly due to the operating companies, who by lending the use of their facilities and personnel, have made this investigation possible. The names of the various power plants have been deleted by request.

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Discussion

Steam-Turbine Leaving Losses and Vacuum Corrections[1]

Ernest L. Robinson.[2] Professor Helander has presented a detailed set of computations of the effect of leaving velocity at the wheel annulus on vacuum corrections for steam-turbine performance. The corrections are based, as they should be, on a particular known performance and the author wisely qualifies his methods as suitable for use in the field where manufacturers' corrections are not available, and his basic theoretical analysis of the steam action in the last-stage bucket exit seems to be correct.

There are, however, a number of influences that must be taken into account in preparing the vacuum corrections for steam-turbine performance. The most important of these is the leaving-velocity loss from the last-stage wheel and this he has considered. Another important element is the pressure drop through the exhaust hood from the wheel annulus to the exhaust flange, which occurs despite considerable excess area. These two influences are closely inter-related and only an approximate analysis of the actual leaving loss can be made without taking account of the variation of pressure due to drop through the hood. Omitting the exhaust-hood loss, however, does give a first approximation to the vacuum corrections since when the exhaust-hood loss is larger, the leaving loss is less. Certain upstream effects are also to be reckoned with for a stage or two preceding the last, particularly at light loads. We understand from the author that these latter effects, while not evaluated, are allowed for in Appendix B if certain factors are known.

Manufacturers' vacuum corrections attempt to evaluate all these influences with the best available knowledge as to the performance of the several elements, supplemented with empirical corrections to the theoretical calculations following analyses of many actual tests. It would be our recommendation that where important decisions are involved, actual tests at various vacuums be carried out or the manufacturers' vacuum corrections, which are based on a wide knowledge of such tests, be used.

C. Harold Berry.[3] The determination of vacuum corrections is an important undertaking in connection with an acceptance test of a steam turbine. The A.S.M.E. Test Code for Steam Turbines now requires that vacuum corrections be established by prior agreement between the manufacturer and the purchaser, either in the purchase contract or in a separate agreement, or else that the corrections be found by auxiliary tests. The first alternative is tantamount to accepting the manufacturer's correction curves; if the purchaser is unwilling to do this, he must conduct a costly series of tests.

Some readers may be of the opinion that Professor Helander has developed another method for securing the desired corrections that will avoid the difficulties of the foregoing alternatives. That this is not the case seems evident from the fact that items 2 to 6 and 28 in the table on page 154 of the original paper are stated to rest upon information furnished by the manufacturer. In the case of a test, item 5 would be known from the test results, and some of the other items might be obtained by sufficiently skillful measurement; but in the end the computed results will rest upon data furnished by the manufacturer and accepted by the purchaser. If the purchaser is willing to accept basic data, he will probably be equally willing to accept the final result. Thus it appears to me that this method should not find a place in the A.S.M.E. Test Codes.

The writer speaks as chairman of Power Test Codes Individual Committee No. 6, which is now engaged in the revision of the Test Code for Steam Turbines. He does not, however, speak for the Committee, since there has been insufficient time for a poll of its membership.

John A. Dent.[4] Both designers and users of turbines will find Professor Helander's paper useful in the estimation of leaving losses and vacuum correction. The influence of critical speed and the limitations of possible vacuum at different loads are clearly brought out. While at first glance the procedure seems rather complicated the development is entirely logical and with a little study and practice can be applied with a reasonable expenditure of time and work.

The problem of determining leaving losses is exceedingly complex, and many variables enter its solution. The author is careful to point out that his method gives only a reasonable approximation and calls attention to some of the neglected variables which may influence the result, namely, leakage, moisture, and variation in nozzle efficiency. Other items which might be mentioned are:

(*a*) Variation in leaving velocity across section of jet
(*b*) Variation in leaving velocity from root to tip of blade
(*c*) Variation in gaging from root to tip.

Under (*a*) it should be remembered that the discharge varies as the mean velocity across the jet, while the kinetic energy varies with the "root mean square" of the velocity. Ordinarily the error in using mean velocity for computing kinetic energy is quite small.

The error in computing kinetic energy from the leaving velocity at mean-blade speed may be considerable in some cases, and entirely negligible in others. For example if the tangential component of the relative velocity is equal to the mean-blade speed, the absolute velocity leaving is a minimum at the middle of the blade, and increases both toward root and tip. If the kinetic energy is calculated on the basis of the velocity at the middle of the blade the result will obviously be too low. In the accompanying diagram, Fig. 1, OA is the relative velocity, AB the mean-blade speed, AC the root-blade speed, and AD the tip-blade speed. Then OB is the absolute velocity at the middle of the blade, OC at the root, and OD at the tip. Plainly, the mean leaving velocity is greater than OB.

The writer made a few calculations on the magnitude of this error using as an example a two-foot blade mounted on a six-foot drum revolving at 1800 rpm. A straight blade was assumed with

[1] Published as paper FSP-57-8, by Linn Helander, in the May, 1935, issue of the A.S.M.E. Transactions.

[2] Turbine Engineering Department, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

[3] Gordon McKay Professor of Mechanical Engineering, The Graduate School of Engineering, Harvard University, Cambridge, Mass. Chairman of A.S.M.E. Power Test Codes Individual Committee No. 6, Steam Turbines.

[4] Professor and Head of Department of Mechanical Engineering, University of Pittsburgh, Pittsburgh, Pa. Mem. A.S.M.E.

a constant gaging giving $\sin \alpha = 0.45$. The figures chosen correspond to a mean-blade speed of 754 ft per sec and a tip speed of 942 ft per sec.

At a relative velocity of 850 ft per sec the kinetic energy was 108 per cent of that found by assuming the leaving velocity corresponding to the mean-blade speed. At 1000 ft per sec the ratio dropped to 103.4 per cent and at 1270 ft per sec became 100 per cent. A maximum of 115 per cent was found at 582 ft per sec.

No attempt has been made to evaluate the effect of change of gaging from root to tip. Such an investigation would require a

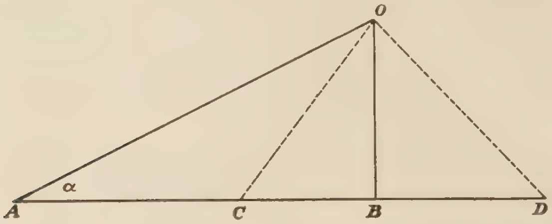

Fig. 1

detailed study of the design of blading used in any individual turbine. Such procedure is far beyond the intended scope of the author's paper. The writer wishes merely to call attention to this factor as one of the variables whose influence has been neglected.

The determination of the critical conditions is one of the most interesting and important contributions in the paper. The curves and approximate formulas developed by the author will do much to save time and labor in this calculation. The data supplied are almost the exact reverse of those given in impulse-turbine nozzles. There a nozzle of given dimensions is supplied with steam of known pressure and quality, and the discharge is easily computed. In the present instance a given discharge is delivered at a known pressure, but unknown quality, and the critical conditions have to be found. A method of successive approximations is used. The author shows that a correct solution is quickly reached. In fact the first approximation appears to be well within the limits of accuracy of the method, but the author properly refuses to take this for granted and carries through the second approximation, showing that his first assumptions were close to the truth.

The writer wishes to express appreciation of the paper and to welcome it as a valuable contribution to our knowledge of the turbine.

Author's Closure

The procedure for evaluating steam-turbine vacuum corrections proposed in the paper was prepared principally for use in making economic and heat-balance studies of the kind usually undertaken before the purchase of turbines and condensers to determine among other data, the vacuum which probably would be the most economical and, therefore, the approximate vacuum for which the turbines and their condensers should be designed.

In view of the uncertainty of data other than vacuum corrections, such as cost data and load curves, which must be estimated or assumed for these preliminary studies, it was felt inadvisable to introduce refinements that might add much to the complexity of the problem of evaluating approximate vacuum corrections and yet have but a small influence on final conclusions. For this reason, and also because precise data on the design of blades and exhaust hoods are not available usually for preliminary studies, the factors to which Professor Dent and Mr. Robinson appropriately call attention were ignored.

Where the necessary data are available and precise results are desired, exhaust-hood losses should be calculated and leaving losses should be more accurately evaluated taking into account, as Professor Dent suggests, variations in gaging from root to tip as well as variations in blade speed from root to tip. When variations in gaging and in blade speed from root to tip are taken into account, the equation for leaving losses takes the form of an integral which can be evaluated graphically if it cannot conveniently be evaluated analytically.

In this connection, it may be well to point out that in evaluating vacuum corrections, two leaving losses calculated for different vacua are subtracted and their errors may be compensatory to some extent. Table 1 of this discussion shows this to be the

TABLE 1 ERROR IN CHANGE IN LEAVING LOSSES ACCOMPANYING A CHANGE IN RELATIVE VELOCITY BASED UPON DATA PRESENTED BY PROFESSOR DENT[4]

Relative velocity, fps	582	850	1000	1100
Approx. leaving losses, Btu per lb	2.48	2.95	4.46	6.00
Error in leaving losses, per cent	15.50	8.00	3.40	1.60
Absolute error in leaving losses, Btu per lb	0.383	0.235	0.151	0.097
Absolute error in change in leaving losses, Btu per lb	Base	0.148	0.232	0.286
	...	Base	0.084	0.138
	...	...	Base	0.054
Error in change in leaving losses, per cent of increment of relative kinetic energy[a]	Base	1.95	1.75	1.65
	...	Base	1.50	1.40
	...	...	Base	1.30

[a] Increment of relative kinetic energy = [(Terminal rel. vel.)^2 — (Base rel. vel.)^2]/50,000.

case for the conditions assumed by Professor Dent. It shows also, for these conditions, that when leaving losses for two different relative velocities are evaluated at the mean diameter, the error in their difference is but a small fraction of the conversion of energy required to accelerate the steam from the lower to the higher velocity, and therefore may be ignored where approximate data are satisfactory.

In the writer's sample calculations, if leaving losses had been evaluated by taking into account variations in blade speed from root to tip on the assumption that straight blades of constant gaging are employed, then the calculated vacuum correction factor, Item 58 on page 155 of the paper, would have changed from 4.60 per cent to 4.67 per cent under Case 1 and from 1.33 per cent to 1.38 per cent under Case 2. For the purposes for which these estimated vacuum corrections are likely to be used, these differences are negligible.

The foregoing considerations do not in any sense detract from the force and value of Professor Dent's criticisms and suggestions. For conditions other than those here investigated the error introduced by evaluating leaving losses at the mean diameter may have a greater effect on the vacuum correction; and as previously stated, where precise values of leaving losses and vacuum corrections are desired, the factors Professor Dent discusses should be considered. The author believes, however, that where the data needed for evaluating leaving losses precisely are not available, his procedure will give approximations of vacuum corrections satisfactory for use in preliminary studies.

The author wishes to thank Professor Dent, Mr. Robinson, and Professor Berry for the discussions they have contributed. Their points of view and criticisms have helped to clarify the intent of the paper and the limitations of the procedure proposed. He agrees with Professor Berry that the procedure outlined for evaluating vacuum corrections should not be made a part of the A.S.M.E. test code.

The Psychrograph[1]

Warren Viessman.[2] The Norris-Westinghouse psychrograph chart is ingenious, and its invention is a material contribution to the design of air-conditioning installations.

The recognition and use of the heat-ratio lines of sensible heat to total heat through a point of known condition, as a locus of all changes of psychrometric conditions of the air under constant-heat ratio (as represented by a point on the chart, and as stated in the first theorem in the paper) offers an accurate and convenient method of determining the saturation temperature of a mixture of air leaving the washer, and greatly simplifies the work of the air-conditioning engineer.

The second theorem given in the paper is useful in establishing the condition of the air entering the washer, and also the condition of the air entering the room, when a by-pass washer with make-up air, as shown in Fig. 1 of this discussion is used. The straight line connecting the two points covering the conditions of the two quantities to be mixed is the locus of all conditions of the mixed air.

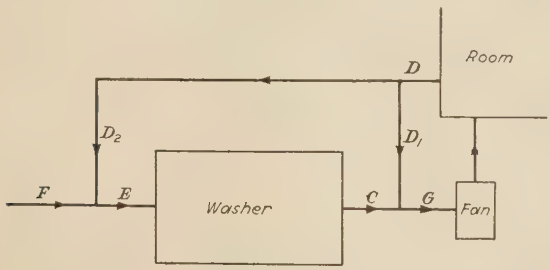

Fig. 1 Diagram of Connections for a By-Pass Washer With Letters Corresponding to Those Given in Example 5. Fig. 5, of the Paper

The chart as constructed is easily read and of much larger scale for the same overall size than the usual chart. The values as read from the chart check to within three significant figures with those obtained from tables and by calculation.

In his Fig. 5, the author has solved a typical by-pass air-washer problem. The letters on the chart in Fig. 5 of the paper, representing conditions at various points, correspond with those on Fig. 1 of this discussion.

In this discussion let C = quantity of air leaving washer, lb per min; D_1 = quantity of room air of condition D recirculating and by-passing the washer, lb per min; D_2 = quantity of room air of condition D recirculating and entering the washer, lb per min; E = quantity of air entering washer to be conditioned, lb per min; F = quantity of outside ventilation air entering washer, lb per min; G = quantity of conditioned and recirculated air entering the room, lb per min; t_c = temperature of air leaving washer, deg F, dry bulb; t_d = temperature of room air recirculated, deg F, dry bulb; t_e = temperature of air entering the washer, deg F, dry bulb; t_f = temperature of outside air, deg F, dry bulb; and t_g = temperature of air entering room, deg F, dry bulb.

The quantity of conditioned and recirculated air G entering the room is determined by the temperature differential desired between the entering air and the room air. The quantity of outside ventilation air F entering the washer is usually expressed as a percentage of G.

[1] Published as paper PRO-57-2, by A. M. Norris, in July, 1935, issue of the A.S.M.E. Transaction.

[2] Associate Mechanical Engineer, Procurement division, U. S. Treasury Department, Washington, D. C. Mem. A.S.M.E.

The quantity of air E entering the washer to be conditioned is

$$E = \frac{\text{Total heat to be removed from room, Btu per hr}}{0.24 \times 60 \text{ (room wet-bulb temp } - t_c)}$$

which is equal to the quantity of air C leaving the air washer, or $C = E$.

The quantity of room air D_1 of condition D recirculating and by-passing the washer is equal to the quantity of conditioned and recirculated air G entering the room minus the quantity of air C leaving the washer, or $D_1 = G - C$.

The quantity of room air D_2 of condition D recirculating and entering the washer is equal to the quantity of air E entering the washer to be conditioned minus the quantity of outside ventilation air F entering the washer, or $D_2 = E - F$. Since $D_1 = G - D$, and $D_2 = E - F$, it is seen that $D_1 + D_2 = G - F$.

The temperature of the air t_c leaving the washer is obtained on the Norris Psychrograph by the intersection of the heat-ratio line through the room condition with the saturation curve, and is 52 F in example 5, Fig. 5, of the paper.

The temperature t_d of the room air recirculated, the temperature t_f of the outside air, and the temperature t_g of the entering air, are selected design temperatures. The temperature t_e of the air entering the washer is equal to $(D_2t_d + t_fF)/E$.

From the foregoing relationships it is seen that the determination of t_c, the temperature of the air leaving the washer, from the Norris Psychrograph readily simplifies a series of relationships which otherwise would be more difficult to solve. The location of t_e and t_g on the locus lines of the chart immediately establishes the condition of the air entering the washer and the air entering the room.

In example 5 given by the author, which is solved in Fig. 5 of the paper, the total heat load of the room was 100,000 Btu for which 59.1 lb of ventilation air was required. The air recirculated through the washer was 118.3 lb per min.

The total heat of outside air F entering the washer at 78 F wet bulb = 40.64 Btu per lb, and the total heat of outside air F leaving the room at t_d deg and 50 per cent relative humidity = 30.80 Btu per lb. Therefore, the heat removed from the ventilation air above room temperature = 40.64 − 30.80 = 9.84 Btu per lb. Hence, the heat removed per hour = 59.1 × 60 × 9.84 = 35,000 Btu per hr. The refrigeration required would then be (100,000 + 35,000)/12,000 = 11.3 tons, which checks with 11.44 obtained by the author in his example 5.

Again, the total heat of recirculated air D_2 to the washer = 30.80 Btu per lb, and the total heat of recirculated air D_2 from the washer at 52 F wet bulb = 21.30 Btu per lb. Therefore, the heat removed from the recirculated air D_2 = 30.80 − 21.30 = 9.50 Btu per lb, and since 118.3 lb per min is circulated, the heat removed per min = 118.3 × 9.50 = 1130 Btu per min. The total heat of outside air F to the washer at 78 F wet bulb = 40.64 Btu per lb, and the total heat of outside air F from the washer at 52 F wet bulb = 21.30 Btu per lb. Therefore, the heat removed from the outside air F = 40.64 − 21.30 = 19.34 Btu per lb, and since 59.1 lb per min pass through the washer, the total heat removed per minute = 59.1 × 19.34 = 1150 Btu per min. The refrigeration required would then be (1130 + 1150)/200 = 11.4, which again checks with 11.44 obtained by the author in his example 5, Fig. 5.

The scale on the Norris Psychrograph giving cubic feet per pound of dry air and vapor present is useful in changing pounds of air to cubic feet for the various conditions when determining fan and duct sizes. The determination of air leakage by use of the Psychrograph will undoubtedly be of service to operating and construction personnel in the locating and the correcting of faults in equipment.

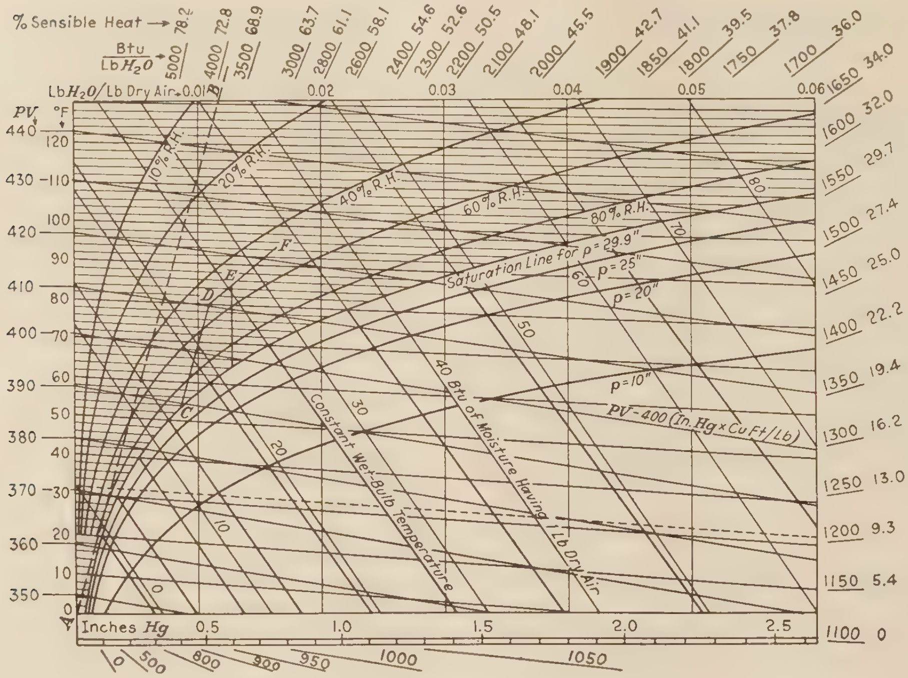

Fig. 2 Mollier Water-Mixture Chart

Arnold Weisselberg.[3] Graphical solutions of air-conditioning problems as recommended by the author are so well established that the merit of such procedure need not be stressed further.

The need for the development of a new chart, such as the author's Psychrograph, for this purpose is open to question in view of the existence of the Mollier water-air-mixture chart which accomplishes the same results in just as simple a manner and is perhaps more convenient to use because it has only one temperature scale compared to three such scales on the author's Psychrograph.

In the Mollier chart the marginal scale furnishes the amount of total heat per pound of moisture removed, which is another way of expressing the ratio of sensible heat to total heat in the author's paper. To make this scale conform exactly to that of the Psychrograph, it is only necessary to divide the heat content of one pound of water at the average temperature of 70 F or 1090 Btu by the scale values and deduct the value of the ratio from unity in order to express the marginal scale in the percentages of the Psychrograph.

[3] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

The procedure is now the same with both charts, except that greater accuracy may be claimed for the Mollier chart, since therein the sensible heat of the vapor in the air is also taken into consideration. This is shown on Fig. 2 of this discussion wherein the writer solves the author's example 5, which in the paper is solved in Fig. 5.

In the second paragraph of theorem 2 on page 229 of the paper, the author states that in case the line joining two points crosses the saturation curve, the locus of possible mixtures will follow the saturation curve instead of crossing it. The word "may" should be used rather than "will" inasmuch as the author himself indicates in the same paragraph that the quality of the mixture will be somewhat uncertain.

In the writer's opinion the chief value of the paper lies in the interesting graphical solution to a variety of problems which are especially adapted to demonstrate the advantages of graphical treatment of an air-conditioning problem. In that respect the author has done an excellent job and the fact that he could have used other charts, which, as has been stated in this discussion, are at least as good for the purpose, appears to be of secondary importance.

Indexes to A.S.M.E. Papers and Publications

THE following pages will serve as a guide to papers and publications of the A.S.M.E. during the calendar year 1935, and also to publications developed by the technical committees. The publications of the Society may be classified in two general groups, regular and special, and were as follows:

REGULAR SOCIETY PUBLICATIONS, 1935

Mechanical Engineering, monthly (see index on pages RI-125–133)
A.S.M.E. Transactions, monthly (see index on pages RI-134–142)
Student Branch Bulletin, published monthly, except June, July, August, and September
Mechanical Catalog, 1935–1936 edition

SPECIAL PUBLICATIONS ISSUED IN 1935

Transactions of the Hydraulic Institute of the Munich Technical University, Bulletin 3 (Authorized translation of Mitteilungen des Hydraulischen Instituts der Technischen Hochschule München, edited by Dr. D. Thoma, Director of the Institute, and published in German in 1929 by Oldenbourg, Munich)
Proceedings of the Fifth Conference of Technical Experts in the Printing Industries (Philadelphia, Pa., October, 1934)
1934 Oil Engine Power Cost Report

Boiler Codes

Revisions and Addenda to Power Boiler Code, Locomotive Boiler Code, Low-Pressure Heating Boiler Code, Miniature Boiler Code, and Unfired Pressure Vessel Code, July, 1935
Low-Pressure Heating Boiler Code, Section IV, A.S.M.E. Boiler Construction Code, September, 1935
Miniature Boiler Code, Section V, A.S.M.E. Boiler Construction Code, August, 1935
Suggested Rules for the Care of Power Boilers, Section VII, A.S.M.E. Boiler Construction Code, April, 1935

Power Test Codes

Centrifugal Compressors and Exhausters, approved May, 1934, published March, 1935
Reciprocating Steam Engines (revision), approved February, 1935, published March, 1935
Instruments and Apparatus
Part 1, General Considerations (revision), approved September, 1935, published November, 1935

Research

Roll Neck Bearings, published June, 1935

Standards

Shafting and Stock Keys (revision), approved October, 1934, published December, 1934
Screw Threads (revision), approved November, 1934, published April, 1935
Jig Bushings, approved October, 1934, published April, 1935
Drawings and Drafting Room Practice, approved February, 1935, published May, 1935
Code for Pressure Piping, approved February, 1935, published, July, 1935
Hose Coupling Screw Threads, approved October, 1934, published July, 1935
Adjusted Pressure Ratings Steel Flanges, approved May, 1935, published September, 1935
Cast Iron Soil Pipe and Fittings, approved August, 1935, published December, 1935
Graphical Symbols, approved April, 1935, published December, 1935
Wrought Iron and Wrought Steel Pipe, approved November, 1934, published December, 1935

Papers Presented at A.S.M.E. Meetings, 1935

The complete technical programs of the meetings of the Society and of its Professional Divisions have been published in *Mechanical Engineering* and may be located by consulting the index on pages RI-125–133. A considerable number of papers and reports included in these programs were not published during the year in either Transactions or *Mechanical Engineering*, but were issued in mimeographed or photo-offset form. Complete sets of these are on file for reference purposes at the office of the Society and the Engineering Societies Library, under the title of "Miscellaneous Papers Presented at A.S.M.E. Meetings, 1935." Photostatic copies of any of the papers may be secured from the Library at regular rates. A list of these papers and reports follows.

Miscellaneous Papers Presented at A.S.M.E. Meetings, 1935

ALMEN, J. O., AND LASZLO, A., The Uniform Section Disc Spring
ANBRO, GOSTA, Power Developments at the Jersey City Plant of Colgate-Palmolive-Peet Co.
BASSETT, PRESTON R., Airport Lighting
BAXLEY, C. H., AND LARSON, C. M., A Survey of Lubricants Used in the Cotton and Woolen Textile Industry
BEATTIE, K. W., Chart for Determination of Water-Hammer Wave Velocities in Cast-Iron Pipe
BEHN, CARL, Experiences With Fuel Injection Equipment in Oil Field Service
BILLINGS, A. W. K., DODKIN, O. H., KNAPP, F., AND SANTOS, A., JR., High-Head Penstock Design
BJORNSSON, C. A., Compressed Air Transmission for Diesel Locomotives
BLAISDELL, A. H., Boundary Layer Flow Over Flat and Concave Surfaces
BOELTER, L. M. K., Insulation and Heat Capacity in Relation to Heat Transfer
BOERLAGE, G. B., AND BROEZE, J. J., Ignition and Combustion of Diesel Fuels
BOSTON, O. W., GILBERT, W. W., AND KRAUS, C. E., The Influence of Cutting Fluids on Tool Life in Turning Steel
BROMBACHER, W. G., Measurement of Altitude Under New FAI Rules
BROWN, C. W., Physiological-Psychological Basis of Comfort
BROWN, RAY, Streamline Tires
BUDDINE, NORMAN T., Heat-Recovery Design for Petroleum Refineries
CAMINEZ, HAROLD, AND BROWN, F. N. M., Large Engine Development
CHERRY, V. H., AND BOELTER, L. M. K., Air Infiltration Through Windows
CONNELLY, JOHN R., AND HERTEL, CHARLES C., Bearing Investigation by the Varying Wear Method
COOKE, MORRIS LLEWELLYN, Power Distribution Costs
COUCH, H. H., Propeller Crankshaft Vibration Problems
CRARY, HAROLD, Air Line Traffic Problems
CROSS, H. C., AND DAHLE, F. B., High-Temperature Properties of Cast and Wrought Carbon Steels From Large Valves for High-Temperature Service
CROSS, H. C., AND DAHLE, F. B., Long-Time Creep Tests of 18 Cr-8 Ni Steel and 0.40 Per Cent Carbon Steel
DALAND, ELLIOT, Rotary Wind Aircraft
DE GANAHL, CARL, Advantages and Possibilities of Shotwelded Stainless Steel in Airplane Construction
DODSON, C. P., The Psychrometric Chart and Its Use
DODGE, WILLIAM G., AND BAUMRUCKER, WM., JR., Newspaper First Impression Printing Problems
DREYER, WALTER, Influence of Water Hammer on Design of High-Head Penstocks at the Drum Plant and Tiger Creek Plant
DUNBAR, HOWARD W., Surface Finishing by Grinding
EATON, GEORGE C., Oil-Engine-Electric Generating Station Operating Costs
ENGER, M. L., Relief Valves and Air Chambers

Foulk, C. W., Suspended Solids in the Foaming and Priming of Boiler Water
Foxwell, W. H., and Woods, B. M., Comfort Cooling Problems and the Rural Home
Freedlander, A. L., Synthetic Resins as a Process Roller Material
Freeman, L. D., Railway Equipment Maintenance
Friess, G. A., New Development in Color Rotogravure Printing
Frisch, Martin, Direct Firing of Pulverized Anthracite Silt
Fritz, L. G., and Hamilton, W. A., Airline Engineering and Maintenance
Gareau, Paul A., Aeronautical Meterology and Scheduled Air Transportation
Geisse, John H., The Small Privately Owned Airplane and Its Probable Future
Girton, Milton D., Airport Management
Glover, Robert E., Computation of Water-Hammer Pressures in Compound Pipes
Goolrick, Robert, Engineering Developments of Army Air Corps
Greene, Carl F., Stressed Skin Aircraft Construction and Its Influence on Other Engineering Construction
Groesbeck, Harry A., Jr., The Photomechanics of Color Photography
Guinther, L. O., Rubber in Aviation
Halmos, Eugene E., The Effect of Surge Tanks on the Magnitude of Water Hammer in Pipe Lines
Harrison, R. E. W., Developments in Modern Drop Forging and Their Relation to Current Machine Shop Practice
Helander, Linn, Steam-Turbine Leaving Losses and Vacuum Corrections, Appendix B, and Appendix C
Hemke, P. E., Radical Developments in Aviation
Hess, Sol, From Design to Type—Processes in the Manufacture of Monotype Matrices
Hooper, O. L., Tests of Pressure Rise Due to Gate Closure at Saluda Hydro Plant
Huebner, William C., Photomechanical Reproduction on Offset Plates
Huebner, William C., Precision Measurement of Color Values
Hunt, R. E., The Printing of Metal Paper
Jewett, A. C., The Graphic Arts Research Bureau
Joachim, W. F., The Characteristics of Diesel Fuel Oil
Johnson, J. B., and Kennedy, R. R., Recent Developments in Metallurgy
Kerr, S. Logan, Water-Hammer Tests in Croton Lake Pumping Plant
Klein, A. L., Review of Aerodynamics for 1934
Kolb, Fred W., Air Filtration
Leland, W. E., Design Procedure for Heating and Cooling Installation
Lincoln, H. L., Cooling With Ice in California Valley Summer Weather
MacPherson, G. R., and Woods, B. M., A Program of Research of Heat Transfer in Unit Coolers
McElroy, John J., Stretch-Out Calculations
McMahan, C. B., Patrolling of Natural Gas Pipe Lines by Airplane
Mertle, J. S., Recent Photomechanical Developments
Mitereff, Sergei D., Supplement of Principles Underlying the Rational Solution of Automatic-Control Problems
Mock, Richard M., Airplane Design and Construction
Moody, Lewis F., Simplified Derivation of Water-Hammer Formula
Myers, David A., Can the Human Body Keep Pace With the Airplane?
Niles, A. S., Effect of Shear Deflection on Axially Loaded Beams
Pagon, W. Watters, Cavitation and Erosion Investigated as a Problem in Fluid Mechanics
Parks, Oliver L., Aviation Schools
Perkins, Kendall, The Air Transport Plane of the Future
Perry, A. S., Study of Effect of Concentrated Sodium Hydroxide on Boiler Steel Under Tension
Perry, Thomas D., Plywood House Units
Peters, O. S., and Buckingham, F., Control of Bearing Pressures in Rolling Mills
Prescott, Ford L., Military Aircraft Engines of the Future
Quick, Ray S., Surge Control in Centrifugal Pump Discharge Lines
Reed, Robert F., Lithographic Research
Rendel, T. B., Fuels for Aircraft Diesel Engines
Rich, Lester J., Analysis of Los Angeles Weather
Richardson, Marion B., Analysis of Occupations of College Grade Compared to the Engineering Professions
Rink, George W., 1934 Progress in Railroad Engineering
Rowe, Albert H., The Medical Aspects of Conditioned Air
Rummel, J. K., and Holmes, J. A., Estimation of Dissolved Solids in Boiler Water by Density Readings
Ruscoe, W. J., Observations on the Technology of Rubber-Plate Manufacture
Savage, W. E., Wing and Tail Flutter, The Causes, Effects, and Remedies
Schroeder, W. C., and Partridge, Everett P., Effect of Solutions on the Endurance of Low Carbon Steel Under Repeated Torsion at 482 F (250 C)
Schwarz, Ralph H., and Winkler, J. Homer, Rubber Plates for Letter-Press Printing
Sellers, Alexander, Jr., How Can Railroad Shops Justify the Use of Modern Machine Tools and Methods?
Smith, P. B., Army Lighter-Than-Air
Spielvogel, S. W., and Kameros, S., Supplement of Application of the "Elastic Point Theory" to Piping Stress Calculations
Stevens, Burt D., New Developments in Offset Printing
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Tillinghast, Harold M., Color Printing in Daily Newspapers
Vander Eb, H. J., Diesel Engine Insurance
Vidal, Eugene L., Aviation Conditions in America
Wagner, G. H., and Wall, J. R., Embrittlement of Boiler Steel by Caustic Soda
Warner, David M., New Methods of Testing Aeronautical Materials and Parts
Washington, Lawrence, The Basic Theory of Air Conditioning
Weeks, Walter S., Analysis of Cooling Fundamentals
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Present Status of Cemented Carbide and Other Modern Cutting Materials
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Comment on Papers in the Water-Hammer Symposium
Report of the A.S.M.E. Committee on Water Hammer
Supplement to the Report of the A.S.M.E. Committee on Water Hammer

Publications Developed by the Technical Committees

The Society's technical committees, the first of which was organized many years ago and all of which have been continuously at work on codes, standards, research and other special reports, have developed a series of publications of permanent value to the membership. The following list is first presented here for record and for ready reference. This list covers the entire group of publications of these committees completed to date which are now available.

To assist the members in securing copies of these publications the sale price is also given. It should be recalled, however, that a discount of 20 per cent is allowed to members of the A.S.M.E. on all pamphlet publications, except standards and the cases specially noted.

RESEARCH

Dynamic Loads on Gear Teeth (1932), $1.50
Determination of Carbonate, Phosphate and Hydroxides in Boiler Waters (1933), $1.75
Fluid Meters
- Part 1—Theory and Application (under revision)
- Part 2—Description of Meters (1931), $1.75
- Part 3—Selection and Installation (1933), $1.50

Tests on Electrical Equipment for Drilling Rotary Drilled Oil Wells (1933), $0.85
Tests on Steam Equipment for Drilling Rotary Drilled Oil Wells (1932), $0.85
Roll-Neck Bearings (1935), $1.50
Bibliography on Cutting of Metals (1866–1930), $1.25
Bibliography on Deterioration of Condensing Equipment (1845–1930), $1.25
Bibliography on Effect of Temperature Upon Properties of Metals (1828–1931), $1.25
Bibliography on Management Literature (1931), $1.75
Supplement to Management Bibliography (in preparation)
Bibliography on Mechanical Springs (1678–1927), $1.25
Bibliography on Woods of the World (1928), $1.25
Steam Tables and Mollier Diagram (1930), $2.50

POWER TEST CODES

Test Codes for

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Centrifugal and Rotary Pumps (1927), $0.50
Compressors and Exhausters (1934), $0.95
Condensing Apparatus (1925), $0.55
Displacement Compressors and Blowers (1925), $0.50
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Supplementary Codes and Publications

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Instruments and Apparatus—
- Part 1—General Consideration (1935), $0.35
- Part 2—Pressure Measurement, Chapter 1, Barometers, and Chapter 6, Tables, Multipliers and Standards (1929), $0.45
- Part 3—Temperature Measurement, Chapter 1, General; Chapter 5, Pyrometric Cones; Chapter 6, Liquids-in-Glass Thermometers; and Chapter 7, Bourdon Tube Thermometers (1931), $0.75
- Part 3—Temperature Measurement, Chapter 8, Optical Pyrometers (1933), $0.40
- Part 4—Head Measuring Apparatus (1933), $0.35
- Part 6—Electrical Measurements (1934), $1.25

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- Part 9—Heat of Combustion (1932), $0.40
- Part 11—Determination of Quality of Steam (1931), $0.45
- Part 12—Measurement of Time (1932), $0.35
- Part 13—Speed Measurements (1930), $0.45
- Part 16—Density Determinations (1931), $0.35
- Part 17—Determination of the Viscosity of Liquids (1931), $0.50
- Part 18—Humidity Determinations (1932), $0.45
- Part 21—Condenser Leakage Tests. Chapter 1 (1928), $0.45
- Part 21—Leakage Measurement, Chapter 2, Boiler and Piping; Chapter 3, Steam Engine Leakage (1932), $0.40

BOILER CODE

Power Boiler Code (1933) Including Materials Specifications and Addenda to 1935, $2.50
A.S.M.E. Unfired Pressure Vessel Code (1934) With 1935 Addenda, $0.75
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Low-Pressure Heating Boiler Code (1935), $0.65
Locomotive Boiler Code (1927) With Addenda to 1935, $0.55
Boiler Code Interpretation Service Data Sheets, $0.15
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Suggested Rules for Care of Power Boilers (1935), $0.70
A.S.M.E. Boiler Construction Code, Combined Edition (1936), $5.50
A.P.I.-A.S.M.E. Rules for Unfired Pressure Vessels for Petroleum Liquids and Gases (1934) (no discount), $1.00

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Index to Mechanical Engineering

Volume 57, January-December, 1935

(BR) denotes Book Review, (C) Correspondence, Ed (Editorial), (S) Survey of Engineering Progress

ADVISORY BOARD OF TECHNOLOGY
Budgeting of effort under (Ed) ... 69

AERIAL TRANSPORTATION
Federal aviation commission recommendation (S) ... 179

AERONAUTICS
A.S.M.E. junior study group on ... 744

AGRICULTURAL ENGINEERING. *See* FARMING

AGRICULTURE
Relation to industry ... 345

AIR
Oxygen for industry from ... 691
Pollution of (S) ... 591

AIR COMPRESSORS
Junkers free-piston (S) ... 246

AIR-CONDITIONING EQUIPMENT
Standards for rating and testing ... 746

AIRCRAFT
Quill bearings for (S) ... 184
Rotary-wing efficiency of (S) ... 244

AIR FILTERS. *See* MINE VENTILATION EQUIPMENT

AIR, FLOW OF. *See* FLOW OF AIR

AIRPLANE CARRIERS
Design of (S) ... 791

AIRPLANE ENGINES. *See also* AUTOMOBILE ENGINES
Automatic power and mixture control for (S) ... 791
European, at Paris Show (S) ... 436
Exhaust tail-pipe for (S) ... 377
Forced-draft cooling for air-jacketed (S) ... 795
Fuel-injection pump for (S) ... 795
Hydrogen as auxiliary fuel in compression-ignition (S) ... 794
Lubrication (S) ... 252
Manufacture, Great Britain (S) ... 245
One-pound-per-horsepower Pegasus X (S) ... 657
Testing parts by magna-flux method (S) ... 446

AIRPLANE MANUFACTURE
Fokker (S) ... 244
Great Britain firm merger (S) ... 244

AIRPLANES
Biplanes (S) ... 584
Gyroplane (Ed) ... 205
Hornet moth (S) ... 584
Landing of (S) ... 112
Lanier "Vacuplane" (S) ... 658
Mignet (S) ... 657
"Pou-du-Ciel" (S) ... 657
"Varivol" (S) ... 658

AIR RESISTANCE. *See* AUTOMOBILE DESIGN, Air-resistance tests

AIR TRANSPORT
Federal Aviation Commission recommendations (S) ... 179

ALCOHOL
Use for fuel (S) ... 251

ALDEN HYDRAULIC LABORATORY
Venturi and weir measurements ... 369

ALFORD, L. P.
Biography of ... 738
Industrial relations (BR) ... 668
Work assignment in silk and rayon manufacturing ... 679

ALLEN, CHARLES M.
Biography of ... 395
Discussion of paper by (C) ... 667
Venturi and weir measurements ... 369

ALLOYS
Beryllium (S) ... 180
Beryllium-copper and beryllium-copper-cobalt (S) ... 659
Heat-resistant (S) ... 717
Iron-chromium-aluminum (S) ... 717
K monel metal (S) ... 659
Low-melting-point (S) ... 585
Tungum (S) ... 658

ALLOY STEEL
Magnetic, new K.S. permanent (S) ... 45

ALUMINUM
Corrosion prevention (S) ... 189

AMERICAN ASSOCIATION FOR THE ADVANCEMENT OF SCIENCE
Officers of elected ... 134
Section M plans for St. Louis Meeting ... 810

AMERICAN BUREAU OF WELDING
Reorganization of ... 674

AMERICAN CHEMICAL SOCIETY
Celebration of American Chemical Industries Tercentenary ... 272
Heat transfer symposium, announcement of ... 812

AMERICAN ENGINEERING COUNCIL
Annual meeting (Ed) ... 70
Annual meeting, report of ... 199
Engineering profession survey ... 132
News notes ... 68, 266, 336, 396, 461, 532, 606, 671, 742, 814
Reese assistant secretary of ... 338

AMERICAN FOUNDRYMEN'S ASSOCIATION
Announcement of 1935 meeting ... 466

AMERICAN GEAR MANUFACTURERS ASSOCIATION
Meeting announcement ... 336
Speed reducer rating adopted by ... 674

AMERICAN INSTITUTE OF ARCHITECTS
S. F. Voorhees president of ... 461

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS
Meeting announcement ... 396
Officers of elected ... 530

AMERICAN INSTITUTE OF MINING AND METALLURGICAL ENGINEERS—A.S.M.E.
Fuel values committee, report of ... 131

AMERICAN PETROLEUM INSTITUTE—A.S.M.E.
Unfired Pressure Vessel Code ... 131

AMERICAN SCIENTIFIC CONGRESS
Mexico City, Sept. 8–17, announcement of ... 606

AMERICAN SOCIETY FOR TESTING MATERIALS
Coming meetings of ... 202
Officers for 1935–1936 ... 464

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
Meeting announcement ... 336

AMERICAN SOCIETY OF MECHANICAL ENGINEERS
Aeronautic division, progress report, copies of available ... 268
Aeronautic meeting (St. Louis, October), announcement of ... 267
Aeronautic meeting (St. Louis, October), papers available ... 816
Aeronautic meeting (St. Louis, October), program for ... 603
Aeronautic meeting (St. Louis, October), report of ... 810
Annual meeting (1934), report of ... 33
Annual meeting (1935) (Ed) ... 678
Annual meeting (1935), announcement of ... 671
Annual meeting (1935), program for ... 738
Annual meeting (1935), technical program plans for ... 604
Applied mechanics meeting (Ann Arbor, June), announcement of ... 267
Applied mechanics meeting (Ann Arbor, June), plans for ... 334
Applied mechanics meeting (Ann Arbor, June), program for ... 395
Applied mechanics meeting (Ann Arbor, June), report of ... 534
Boiler code interpretations. *See* BOILER CODE
Business meeting (June, 1935), report of ... 511
Business meeting (December, 1934), report of ... 37
Candidates for membership ... 68, 136, 204, 272, 340, 400, 466, 536, 608, 676, 746, 816
Capital-goods industries committee, report of ... 9
Committee appointments ... 461
Committee on manual of practice, cooperation with (Ed) ... 402
Constitution, proposed amendments to ... 532
Constitution, passage of amendments to ... 814
Council executive committee meetings, reports of ... 63, 200, 395, 744
Council meeting (June, 1935), report of ... 530
Council meeting (October, 1935), report of ... 808
Council meeting (December, 1934), report of ... 37, 38, 62
Employment, committee on, report of ... 129
Estimated cost of activities for 1935–1936 (table) ... 531
Estimated income for 1935–1936 (table) ... 531
Fifty-year badges, presentation of ... 36, 265
Financial condition ... 331

AMERICAN SOCIETY OF MECHANICAL ENGINEERS (*continued*)
Graphic arts division formed ... 267
Honorary membership, conferring of ... 34
Junior aeronautic study group ... 744
Junior award, conferring of ... 34
Junior group, second anniversary meeting of ... 39
Junior participation in affairs of ... 608
Local sections, coming meetings of ... 68, 136, 202, 268, 340, 746, 816
Lubrication engineering committee, reorganization of ... 132
Machine shop sessions at Machine Tool Congress (Cleveland, September), program for ... 534
Main award, conferring of ... 37
Management division committee chairman ... 134
Manual of practice committee, report of ... 130
Medal, presentation of ... 36
Members, deaths of ... 68, 136, 204, 272, 340, 466, 536, 608, 676, 746, 816
Members, memorial notices of (Ed) ... 537
Members, professional data on (Ed) ... 273
New secretary of ... (Ed) 1, 38
Nominating committee for 1935–1936 ... 38
Nominating committee solicits suggestions ... 134
Officers for 1935-1936, election of ... 744
Officers for 1935–1936, nominations for ... 527
Oil and gas power meeting (Tulsa, May), announcement of ... 268
Oil and gas power meeting (Tulsa, May), program for ... 334
Oil and gas power meeting (Tulsa, May), report of ... 461
Petroleum division, chairman of appointed ... 134
Printing industries division changes name ... 267
Process industries division appointments ... 268
Professional divisions, progress reports of, 1934 ... 99
Railroad division progress report, copies of available ... 268
Reemployment program ... 129
Semi-annual meeting (1935), announcement of ... 267
Semi-annual meeting (1935), papers requested for ... 134
Semi-annual meeting (1935), plans for ... 333, 390
Semi-annual meeting (1935), program for ... 391
Semi-annual meeting (1935), report of ... 509
Student award, conferring of ... 37
Student branch bulletin (Ed) ... 609
Technical committee meetings (December, 1934), report of ... 40
Textile meeting (Greenville, April), announcement of ... 267
Transactions, papers in ... 68, 136, 204, 272, 340, 400, 466, 536, 608, 676, 746, 816

A.S.M.E.—A.I.M.E.
Fuel values committee, report of ... 131

A.S.M.E.—A.P.I.
Unfired Pressure Vessel Code ... 131

A.S.M.E.—A.S.C.E.
Hydraulic meeting, program for ... 66

ANDERSON, RUSSELL H.
Biography of ... 670
Technical ancestry of grain-milling devices ... 611

ANGUS, ROBERT W.
Biography of ... 808
Kreitner's diagram for water-hammer problems ... 781

APPLIED MECHANICS
Fourth international congress, proceedings of ready ... 66
Stresses in overstrained materials (S) ... 791

APPRENTICES
Training, methods of ... 421
Training, national ... 420

ARMOUR, JAMES W.
Slag tapping of boiler furnaces (C) ... 667

ARMSTRONG, FLOYD E.
Biography of ... 808
Security markets ... 766

ARMY ORDNANCE ASSOCIATION
A.S.M.E. members invited to meeting of ... 672

ARNSTEIN, KARL
Biography of ... 527, 603
"The Comet"—high-speed train ... 479, 553

ATOMIZATION. *See* DIESEL ENGINES, Fuel injection

ATTERBURY, WILLIAM WALLACE
Obituary of 789
Portrait of 789

AUTHORS
Biographical sketches of (Ed) 137

AUTOMOBILE DESIGN
Air resistance tests (S) 442

AUTOMOBILE ENGINES
Airplanes, use in, with belt drive (S) 794
Waukesha (S) 251

AUTOMOBILE GEARS. *See* AUTOMOBILE TRANSMISSIONS

AUTOMOBILE INDUSTRY
Influence of (Ed) 747

AUTOMOBILES
Features of 1935 models 78

AUTOMOBILE TRANSMISSIONS
Automatic (S) 51
de Normanville (S) 720

AUTOMOTIVE FUELS
Liquefied gases (S) 380
Petrogas (S) 185
Research on (S) 384
Wood-coal producer gas (S) 51

AVIATION
Blind flying (Ed) 537

B

BAILEY, ALEX D.
Biography of 527
Photograph of 42, 529

BALTIMORE ENGINEERS CLUB
Engineering addresses at 87, 174, 237, 277, 364

BATT, WILLIAM L.
Biography of 527
Henry Laurence Gantt (BR) 522
Photography of 529

BAUMEISTER, THEODORE, JR.
Biography of 527
New curriculum in mechanical engineering 483

BEAMS
Bending, theory of (S) 113

BEARING METALS
Lead-base (S) 180
Tests on cast iron as (S) 584

BEARINGS
Lubrication 292
Surface corrosion 239

BELTS
Textile, coefficient of friction with pulley materials (S) 719

BENNETT, JOSEPH S.
Biography of 808
Design and development of water-cooled underfeed stoker 761

BENSON, HARRY
Biography of 670
Effective handling of quality complaints. 636

BERRY, C. HAROLD
Biography of 670, 738
Steam-turbine testing 705
Thermodynamic properties of compressed liquid water 618

BERRY, F. E., JR.
Cost of noise (C) 732

BERYLLIUM
Alloys (S) 180

BERYLLIUM ALLOYS
Copper and copper-cobalt (S) 659

BESSEL FUNCTIONS
Types of (S) 49

BIBLIOGRAPHIES
Available at library 398
Cavitation research 216
Dimensional control 780
Heat transmission 704
Metal cutting, publication announced 672
Occupational diseases 234
Photoelasticity 771
Quality control 642
Steam pressure measurement 635

BIERBAUM, CHRISTOPHER H.
Biography of 265
Corrosion of bearing surfaces 239

BILLING, PEMBERTON
Durotofin invented by (S) 179

BILLITER, J.
Solid smokeless fuel (C) 388

BILLMYER, CARROLL D.
Comments on E.C.P.D. (C) 596

BINARY VAPOR CYCLES
Diphenyl and Findlay (S) 118

BIOMECHANICAL REPERCUSSIONS
Editorial on 402

BIOMECHANICS
Editorial on 205

BLACKBERG, SOLON N.
Biography of 265
Engineering methods in medical research. 217

BLACKWELDER, C. D.
Biography of 670
Stream pollution and textile wastes 631

BLADES. *See* STEAM TURBINES, Blades

BOEGEHOLD, A. L.
Modern cast iron (C) 595

BOILER CODE
Boiler construction code, revisions and addenda to 126, 260, 324, 387
Interpretations 56, 125, 194, 260, 387, 526, 598, 733

BOILER CORROSION
Prevention with sodium sulphite (S) 315

BOILER FEEDWATER. *See* FEEDWATER

BOILER FURNACES
Slag tapping of 367

BOILER PLATES
Cracking 294

BOILERS. *See also* BOILERS, HIGH-PRESSURE
Electric. *See* BOILERS, ELECTRIC
Embrittlement, caustic (S) 384
High gas velocity .. (S)380, (S)445, 469, (S)793
High-pressure. *See* BOILERS, HIGH-PRESSURE
Horizontal thermax (S) 520
Self-contained economic, heat loss with (S) 253
Single-tube or flash (S) 186
Steam. *See* STEAM BOILERS
Thimble-tube, heat transmission in (S) .. 319
Velox .... (S)254, (S)380, (S)445, 469, (S)793

BOILERS, ELECTRIC
Brown Boveri (S) 444

BOILERS, HIGH-PRESSURE
Fusion-welded drums for (S) 521
Loeffler (S) 186
Welded drums for (S) 799

BOILERS, STEAM. *See* STEAM BOILERS

BOILER STEEL
Embrittlement 294

BOILERS, WATER-TUBE
Wagner-Bauer (S) 441

BOLTS
Impact and static tensile properties of (S) 517

BOND, GEORGE M.
Tribute to (Ed) 71

BOOKS RECEIVED 59, 128, 197, 263, 329, 389, 456, 525, 601, 669, 735, 805

BOOKS REVIEWED
America's capacity to consume 507
America's capacity to produce 507
America's output and income (Ed) 467
Art of leadership 262
Autobiography of John Hays Hammond. 599
Balance of international payments of United States 714
Controlling depressions 376
Corporate profits as shown by audit reports 601
Correct economy for the machine age .... 235, (C)451
Diesel engines 805
Editorial 206
Electrons (+ and −), protons, photons, neutrons, and cosmic rays 522
Executive guidance of industrial relations 668
Flexible budget 389
Formation of capital 507
Heat calculations for design and operation of furnaces 735
Henry Laurence Gantt: Leader in industry 522
Identification of firearms 803
Industrial profits in the United States 195
Industrial standardization 329
Internal-combustion engine 262
Internal debt of United States 582
Inventing the ship 734
Isaac Newton—A biography 524
Liquid claims and national wealth 582
Marketing industrial equipment 327
Mathematical treatise on vibrations in railway bridges 58
Mathematics essential for elementary statistics 456
Mechanization in industry 303
Modern housing 647
Nature of capitalist crisis 434
New pathways in science 327
Parsons, Charles A., scientific papers and addresses of (Ed) 273
Psychometrics 328
Report of the investigation of engineering education, 1923–1929 262

BOOKS REVIEWED (*continued*)
Research—pathfinder of science and industry 600
Road and rail in forty countries 454
Security markets 766
Sociology of invention 734
Theory of wages 375
Trend analysis of statistics, theory, and technique 456
"Verein Schweizerischer Maschinen-Industrieller—1883–1933" 196

BOTTLE MAKERS
Karapetoff parable of (C) 124

BOULDER DAM
Turbines for 539

BRADLEY, FRANK L.
Biography of 395
Flue-gas scrubbers 361

BRASS MILLS
Canada (S) 188

BRASTIL
Properties of (S) 46

BRAZING
Silver solders (S) 313

BRIGMAN, B. M.
Appointment to Louisville Commission .. 466

BROWN, THEODORE H.
Control of distribution (C) 258

BUCKINGHAM, EARLE
Industrial standardization (BR) 329

BUILDINGS
Heat insulation by galvanized corrugated-iron sheathing 687

BURDELL, EDWIN S.
Biography of 670
Discussion of book review by (C) 733
Engineer and housing problem (BR) 647

BUREAU OF LABOR STATISTICS SURVEY
Explanation of 536

BURRELL, B. S.
Biography of 459
Discussion of paper by (C) 594
Steel-plant lubrication 415

BUTTS, HARRY F.
Identification of firearms (BR) 803

C

CALIFORNIA, UNIVERSITY OF
Diesel-engine course announced by 672

CAMMEN, LEON
Swiss machinery industry (BR) 196

CAPILLARITY
Coal (S) 793

CAPITAL AND CREDIT
Control of 20

CAPITAL-GOODS INDUSTRIES. *See also* DURABLE-GOODS INDUSTRIES
Code restrictions in, report on 9

CARLIN, P. H.
"Civil Engineering" editor 396

CARMAN, GEORGE NOBLE
Retires as director of Lewis Institute .. 272

CARNEGIE, ANDREW
One-hundredth anniversary of (Ed) 747

CARRIER, WILLIS H.
A.S.M.E. Medal presented to 36
Portrait of 36

CARS, FREIGHT
Baltimore & Ohio (S) 188

CARTER, GEORGE W.
Biography of 333
Discussion on paper by 388
Solid smokeless fuel 305

CASEY, HUGH J.
Passamaquoddy tidal-power project 580

CASTING. *See* FOUNDRY PRACTICE

CASTINGS
Automobile, German progress in (S) 717
Steel. *See* STEEL CASTINGS

CAST IRON
Bearing metal tests (S) 584
Modern 429
Transverse and tensile tests of (S) 792

CAVITATION
Research on 211

CENTENARIES, ENGINEERING
1935 (Ed) 142

CHARLES T. MAIN AWARD
Presentation of 37

CHEMICAL INDUSTRIES
England, reorganization of (S) 448
Exposition of, announcement of 676
Exposition of, plans for 810

CHENEY, HORACE B.
Gantt medal presented to.............. 39
Portrait of.......................... 40
CHRISTIE, A. G.
Biography of......................... 134
Engineers' business contacts.......... 87
CHROMIUM PLATING
Effect on health of workers (table)...... 231
CINEMATOGRAPHY
Rifle fire research by (S)............... 379
Textile machinery research by.......... 446
Use in research............(Ed) 341, (S) 446
CITIZENS
Engineers as......................... 209
CIVILIZATION
Progress of.......................... 5
CLARK, WALLACE
Gantt medal presented to.............. 39
Portrait of.......................... 39
COAL
Byproducts from (S).................. 586
Cannock system of oil distillation from (S)........................... 113
Capillarity of (S)..................... 793
Cleaning by gravity (S)................ 249
Fluorine in (S)....................... 113
Spontaneous oxidation of (S)........... 249
COAL DUST. *See* ENGINE FUELS, Coal dust
COAL PULVERIZERS
Buell Pulvez mill (S).................. 52
COAL-TAR OILS. *See* ENGINE FUELS, Coal-tar oils
CODES. *See* BOILER CODE; UNFIRED PRESSURE VESSELS, A.P.I.-A.S.M.E. Code
COLEMAN, WILLIAM B.
Biography of......................... 333
Frederick Walton..................... 297
COLLIER TROPHY
Presentation to Hegenberger (Ed)....... 537
"COMET"
High-speed train.................. 479, 553
COMPRESSED AIR
Receiver explosions (S)................ 180
COMPRESSORS
Rotary, Surth single-stage (S).......... 377
COMSTOCK, CHARLES W.
Comments on E.C.P.D. (C)............ 450
CONDENSER TUBES
Recent developments in................ 628
CONDUITS
Coefficient of roughness under pressure (S).............................. 718
CONSUMERS-GOODS-INDUSTRY
Employment in.............. 198, 335, 395
CONVECTION. *See* HEAT LOSSES
COOK, H. M.
Suggestion systems (C)................ 194
COOLEY, MORTIMER E.
A.S.M.E. Fifty-Year badge presented to. 265
COOPERATIVE SYSTEM
Backgrounds of....................... 418
COPPER
Drilling (S).......................... 662
COPPER ALLOYS
Brastil (S)........................... 46
COPPOCK, WALTER J.
Pivoted-motor drive (C)............... 593
CORNELL UNIVERSITY
Large-scale planning courses........... 66
CORROSION. *See also* METALS CORROSION
Aluminum (S)......................... 189
Bearing surfaces...................... 239
Boiler. *See* BOILER CORROSION
Metals. *See* METALS CORROSION
COTTON SPINNING
Casablancas' compound drafting machine (S)............................ 53
CRANKCASE OIL
Viscosity of Diesel (S)................. 381
CRANKSHAFTS
Local surface hardening of (S).......... 517
CROFT, HUBER O.
Biography of......................... 459
Discussion of paper by (C)............. 597
Engineer and his societies.............. 431
CROSS, R. C.
Oil burners and boilers (C)............. 453
CULLIMORE, ALLAN R.
Biography of......................... 265
Engineer as a citizen.................. 209
CUTTING OF METALS
Built-up edge, formation and function of. 487
Structural steel....................... 423
CUTTING TOOLS
Speed-life curves for (S)............... 718
CYCLOPENTADIENE
Extraction from coke-oven tars (S)...... 586

D

DAGGETT, PARKER H.
Biography of......................... 808
Engineer registration laws............. 783
DALLAVALLE, JOSEPH M.
Biography of......................... 265
Prevention of occupational diseases...... 230
DAMS. *See also* BOULDER DAM
Wheeler, turbine for................... 499
DAVANNE MEDAL
Award to Epstean..................... 816
DAVIES, C. E.
Portrait of.......................... 38
The new secretary!............... (Ed) 1, 38
Visits to Local Sections................ 202
DEBAUFRE, W. L.
Atmospheric air...................... 691
Biography of......................... 738
DESSION, GEORGE H.
Identification of firearms (BR).......... 803
DETONATION. *See* INTERNAL COMBUSTION ENGINES, Detonation
DICKINSON, Z. C.
Discussion of paper by............... (C) 122, (C) 192, (C) 321
DICKSON, JOHN
Biography of......................... 603
Diesel engine for rail transportation..... 547
DIE CASTING
Pressure (S).......................... 792
DIE CASTINGS
Brastil copper alloy for (S)............. 46
Copper-alloy (S)...................... 256
DIESEL ENGINES
Airless, conversion to air-injection (S)... 47
Archaouloff-Krupp (S)................. 47
Automotive, Russian (S)............... 797
Coal-dust......... (S) 182, (Ed) 207, (S) 250
See also Pawlikowski
Fuel injection (S)..................... 183
Growth of the industry in America (S)... 250
Large (S)............................ 315
Pawlikowski (S)...................... 250
Rail transportation, use for............ 547
Small (S)............................ 311
Small four-cycle (S)................... 438
Statistics on use in Germany (S)........ 518
Use for pump drives (S)............... 445
Witte (S)............................ 381
DIESEL LOCOMOTIVES
Building industry use (S).............. 254
DIETZ, ALBERT
Financial aspects of housing (abstract).. 572
DIMENSIONS
Control in manufacturing.............. 772
DIPHENYL
Use in heat engines (S)................ 118
DISASTERS
Lessons from (Ed)..................... 2
DISEASES
Occupational, prevention of... 90, 92, 95, 230
DISTRIBUTION
Control of (C)........................ 257
DISTRICT HEATING
Zurich (S)........................... 117
DITHRIDGE, E. B.
Suggestion systems (C)................ 192
DODDS, W. M.
Biography of......................... 738
Underfeed-stoker practice in Great Britain........................... 695
DODGE, H. F.
Biography of......................... 670
Statistical aspects of sampling inspection plans................................ 645
DOMES
McDonald Observatory....... (Ed) 341, 349
DORCUS, ROY M.
Biography of......................... 332
Personality evaluation and business relations................................ 277
DREWRY, M. K.
Application of central-station practice to domestic heating.................... 103
Biography of......................... 134
Domestic heating (C).................. 259
DRILLS
Portable electric, maintenance costs of (S) 720
DRYING EQUIPMENT
Testing method proposed.............. 202
DUCTILITY. *See* METALS TESTING, Ductility
DUDLEY, WILLIAM LYLE
Biography of......................... 528
Photograph of........................ 529
DURABLE-GOODS INDUSTRIES. *See also* CAPITAL-GOODS INDUSTRIES
Code restrictions in, reports on.. (Ed) 2, (Ed) 3
Fundamentals for recovery in.......... 15
Problems confronting.................. 22
DURAND, WILLIAM F.
Guggenheim Medal awarded to......... 396
Honorary membership conferred upon... 34
Isaac Newton (BR).................... 524
John Fritz medal awarded to........... 814
Portrait of.......................... 34
DUROTOFIN
Description of (S)..................... 179
DUST
Control of........................... 154
Examination of, by polarized light (S).... 725
Toxic................................ 95
DUST COLLECTORS
Improved design for................... 753

E

EASTMAN KODAK COMPANY
Lubrication methods.................. 565
ECONOMIC PLANNING
National............................. 15
ECONOMIC SECURITY
Probability of (Ed).................... 677
ECONOMICS
Engineering study of (S)............... 591
Reviews of books on .. (Ed) 206, 235, (C) 451
ECONOMIZERS
Ribbed-tube for horizontal gas flow (S).. 187
EDGERTON, HAROLD E.
Biography of......................... 265
Study of flow of air with stroboscope.... 228
EDISON MEDAL
Awarded to Willis R. Whitney.......... 132
EDISON, THOMAS ALVA
Memorial to announced................ 814
EDUCATION. *See* ENGINEERING EDUCATION; INDUSTRIAL EDUCATION
EDWARDS, L. N.
Study of technical history (C).......... 449
EIDMAN, F. L.
Biography of......................... 527
New curriculum in mechanical engineering 483
ELECTRIC DRIVE
Pivoted-motor........................ 287
ELECTRIC MOTORS
Thyratron (S)......................... 45
ELECTRIC ROTORS
Forgings, testing of (S)................ 115
Hysteretic influences on whirling of (S).. 716
ELECTRIC UNITS
Edmmus system (S)................... 446
ELOXAL PROCESS
Corrosion prevention (S)............... 189
EMBRITTLEMENT
Boilers, caustic (S).................... 384
EMPLOYMENT
A.S.M.E. committee on................ 129
Engineers (C)........................ 258
Junior member study of opportunities for (Ed).................. 137, 198, 335, 395
ENGINEERING
Business fundamentals in.............. 87
Commercial aspects of................. 174
Medical research and......... 217, (Ed) 402
ENGINEERING EDUCATION. *See also* INDUSTRIAL Education
Columbia plan........................ 483
Cooperative system................... 418
Developments in (Ed).................. 468
ENGINEERING FOUNDATION
Annual report of...................... 335
News of............................. 336
ENGINEERING HISTORY. *See* HISTORY, ENGINEERING
ENGINEERING INDEX, INC.
Solicitation of funds by............ 268, 464
ENGINEERING MECHANICS
Summer Session at Michigan announced. 270
ENGINEERING PROFESSION
Labor Department survey of........... 132
Recovery in (Ed)..................... 1
Survey of............................ 132
ENGINEERING PROGRESS..... 43, 110, 179, 244, 309, 377, 436, 513, 584, 657, 716, 791
ENGINEERING SOCIETIES
Membership in, advantages of (Ed)..... 341
Organization of............. (Ed) 401, 431
ENGINEERING SOCIETIES EMPLOYMENT SERVICE
Men available registered with.......... 136

Engineering Societies Library
Bibliographies available at ... 398
Report for 1934 ... 270
Service bureau of ... 674
Engineers
Advertising for (Ed) ... 401
Business contacts of ... 87
Economics and (S) ... 591
Employment (C) ... 258
English of ... 364
Metallurgists and ... 216
Personal qualifications of ... 237
Public life, place in ... 209, (Ed) 748
Registration laws for, E.C.P.D. report on 783
Social training (Ed) ... 205
Engineers Club of Baltimore
Engineering addresses at ... 87, 174, 237, 277, 364
Engineers' Council for Professional Development
Annual report of, 1935 ... 809
Certification program of ... 461
Comments on (C) ... 450
Engineering schools accredited by ... 534
Engineering schools ask to be accredited by ... 676
Engineering schools, report of committee on ... 66
Program described to engineers in New York ... 266
Report on engineer registration laws ... 783
Scott elected chairman of ... 742
Seabury elected secretary of ... 130
Second annual report of ... 338
Engineers' National Relief Fund
Report of ... 270
Engine Fuels
Coal dust ... (S) 182, (Ed) 207, (S) 250
Coal-tar oils (S) ... 182
Engines
See also Diesel Engines; Internal Combustion Engines; Steam Engines; Turbines
Airplane. *See* Airplane Engines
Turbulence-chamber (S) ... 438
Engines, Marine. *See* Steam Engines, Marine
Epstean, Edward
Davanne Medal awarded to ... 816
Ernst, Hans
Biography of ... 527
Metal cutting ... 487
Ethylene
Synthetic lubricating oil from (S) ... 382
Everett, Franklin L.
Biography of ... 204
Stress analysis of failures in machine parts 157
Explosions
Air-compressor receivers (S) ... 180
Extrusion
Flow of metal during (S) ... 255

F

Fairley, Lincoln
Biography of ... 395
Labor's dilemma in depression ... 375
Falk, Harold S.
Discussion of paper by (C) ... 597
Farmer, P. M.
"A misleading statement" (C) ... 191
Farming
Mechanization of (S) ... 513
Fatigue
Failures due to, in machine parts ... 157
Fatigue of Metals
Notch sensitiveness (S) ... 309
Federal Government
Expenditures and income ... 347
Feedwater Treatment
Colloidal (S) ... 723
Feely, F. J.
Biography of ... 670
Discussion of paper by ... 643, 645
Quality control in manufacture ... 638
Fellows, C. H.
Appointment as chairman of boiler feed-water studies ... 461
Ferris, Walter
Machine-shop practice (C) ... 124
Fiber
Machining (S) ... 48
Findlay Cycle
Binary vapor (S) ... 118
Finley, John H.
Rice memorial lecture ... 36
Fires
Oil, extinguishing (S) ... 315
Flanders, Ralph E.
Biography of ... 395
Neglected elements of recovery ... 345
Portrait of ... 33
Visits to Local Sections ... 202
Warner Medal presented to ... 36
Flanges
Insulation of (S) ... 663
Flow of Air
Stroboscopic study of ... 228
Flow of Fluids
Turbulence ... 407
Flow of Metal. *See* Extrusion
Fluorine
Presence in coal (S) ... 113
Fogs
Dispelling (Ed) ... 69
Forging Machines
Solid-frame (S) ... 520
Forgings
Flakes in (S) ... 185
Forstmann Woolen Company
Flue gas scrubbers ... 361
Founder Societies. *See* National Engineering Societies
Foundry Equipment
Sand Wizard shot-blast cleaning machine (S) ... 659
Foundry Practice
Die casting under pressure (S) ... 792
Multi-casting process (S) ... 660
Freeman, Harold A.
Biography of ... 459
Foundations of Marxism (BR) ... 434
Freeman, Ralph E.
Biography of ... 265, 738
Correcting our economy ... 235
Our international accounts (BR) ... 714
Freeman Scholarship
Applications for ... 200
Freight Cars
Light (S) ... 188
French, John C.
Biography of ... 395
Engineer's English ... 364
Frequency
Measurement of ... 215
Freund, Clement J.
National apprenticeship (C) ... 597
Fuel Engineering
Research, British (S) ... 181
Fuel Injection
Airplane engines, pump for (S) ... 795
Fuel Oils
Cracked-residue, burning of ... 241
Fuels
Alcohol (S) ... 251
Developments in, 1934 ... 99
Gasoline engine (S) ... 514
Solid smokeless ... 305
Fuel Oils
Combustion products of (S) ... 661
Fuel Values
Committee report on ... 131
Fullmer, I. H.
Biography of ... 808
Fundamentals of mechanical dimensional control ... 772
Furnaces, Boiler. *See* Boiler Furnaces

G

Gaging
Principles of ... 777
Gantt Medal
Award to Young ... 396
Presentation of ... 39
Garrett, W. S.
Domestic heating (C) ... 258
Gases
Furnace, waste-heat from (S) ... 383
Physical and toxic properties of common (table) ... 232
Gasoline Engines
Fuels for (S) ... 514
Gas producer for (S) ... 46
Gasolines
Vapor pressure and heat of vaporization of (S) ... 514
Gas Plants
Elimination of (Ed) ... 138
Gas Producers
Guylin charcoal (S) ... 46
Gates, Robert M.
Marketing industrial equipment (BR) ... 327
Gears. *See also* Worm Gears
Automobile. *See* Automobile Transmissions
Helical and herringbone, horsepower capacities of (S) ... 48
Helical tooth, deflection of ... 225
Worm reduction, efficiency formula for (S) ... 661
Gear Teeth
Forged (S) ... 795
Helical, deflection of ... 225
Gerry, Harold T.
Equation of state for superheated steam. 164
Gibbons, James O. G.
Wine makers—another view (C) ... 191
Gibson, Geo. H.
Socialization of housing (C) ... 733
Gilbert, E. M.
Biography of ... 459
Steam versus hydro power ... 403
Gilbreth, Lillian
Appointed professor at Purdue University 398
Psychometrics (BR) ... 328
Suggestion systems (C) ... 323
Gill, Augustus H.
Are metals permeable by oil? ... 413
Biography of ... 459
Ginnings, D. C.
Steam research at the Bureau of Standards ... 162
Gliders
Auxiliary sailplane (S) ... 716
Maikafer (S) ... 513
Grain Milling
Development of devices for ... 611
Graphic Arts Research Bureau
First official meeting of ... 510
Graphic Charts
Time-series ... 814
Gratz, Charles Murray
Biography of ... 265
Engineering methods in medical research. 217
Work of (Ed) ... 205, 402
Gray, Albert S.
Administration of occupational disease control ... 92
Biography of ... 134
Guggenheim Medal
Award to Durand ... 396
Guns
Double-barrel small-caliber (Ed) ... 402
Guylin Gas Producer
Use for gasoline engines (S) ... 46
Gyroplanes
Kay ... (Ed) 205, (S) 244
Wilford (S) ... 716

H

Hamerstadt, W. D.
Pivoted-motor drive (C) ... 594
Hamilton, C. S.
Suggestion systems (C) ... 123
Hanlon, M. J.
Biography of ... 265
Burning of cracked-residue fuel oils ... 241
Hardgrove, R. M.
Slag tapping of boiler furnaces (C) ... 667
Harden, John D.
Suggestion systems (C) ... 193
Harrison, Harry
President of A.S.R.E. ... 64
Harrison, R. E. W.
Correction for paper by ... 66
Discussion of paper by (C) ... 124
Hartford, Ernest
Southern itinerary ... 268
Hatch, Theodore
Biography of ... 204
Dust control ... 154
Heat Engines
Diphenyl (S) ... 118
Heating
Domestic (C) ... 258
Domestic, application of central-station practice ... 103
Velox hot-water boilers for (S) ... 793
Heating Systems
Oil burners and boilers ... 221, (C) 453

Heat Insulating Materials
Metal sheets.... 621
Heat Insulation
Buildings. *See* Buildings, Heat insulation
Heat Losses. *See also* Power Plants, Heat losses in
Reduction and convection, separation of. 700
Heat-Transfer
Classification proposed.... 744
Drop condensation and (S).... 53
Symposium on announced.... 812
Heat Transmission
Boilers and water heaters (S).... 319
Hedrick, E. R.
New physics (BR).... 522
Hegenberger, Albert F.
Collier Trophy presented to (Ed).... 537
Hencky, H.
Biography of.... 204
Stresses in rubber tires.... 149
Herbeck, Charles J.
Biography of.... 808
Design and development of water-cooled under-feed stoker.... 761
Higgins, N. B.
Turbines for Wheeler Dam (C).... 727
High Temperatures
New means of producing (S).... 256
Hirshfeld, C. F.
Biography of.... 134
Comments on E.C.P.D. (C).... 451
Progress in power.... 99
History, Engineering
Growing interest in (Ed).... 137
Study of....143, (C) 449
Hoadley Franklin R.
Biography of.... 68
Problems confronting durable-goods industries.... 22
Hodson, W. D.
Steel-plant lubrication (C).... 594
Holley Medal
Presentation to Irving Langmuir.... 510
Hood, O. P.
Biography of.... 738
Reflections.... 685
Hooper, Leslie J.
Biography of.... 395
Discussion of paper by (C).... 667
Venturi and weir measurements.... 369
Hoover, H. M.
Suggestion systems (C).... 193
Hopkins, W. E.
Biography of.... 66
Code restrictions on machinery and production.... 9
Houses
Heating (C).... 258
Heating, oil burners for....221, (C) 453
Heating systems for.... 103
Prefabricated, abstracts of papers on.... 571
Housing
Engineer's relation to problem of (BR).. 647
Literature on (Ed).... 609
Prefabricated.... 571
Houston, George H.
Biography of.... 68
Fundamentals for recovery in durable-goods industries.... 15
Hunsaker, Jerome C.
Biography of.... 265
Cavitation research.... 211
Hunter, John A.
Biography of.... 528
Photograph of.... 529
Huttner Steam Turbines. *See* Steam Turbines, Huttner
Hydraulic Laboratories
Alden, Worcester Polytechnic Institute.. 369
Hydraulic Power
Progress in, 1934.... 99
Hydraulics
Munich reports on.... 812
Hydraulic Turbines
Kaplan automatic adjustable-blade (S).. 794
Hydrodynamics
Turbulence.... 407
Hydroelectric Power
Steam vs.... 403
Hydroelectric Power Plants
Dnieprostroy.... 281
Hydrogen
Heavy.... 141

I

Industrial Education. *See also* Engineering Education
National.... 420
Industrial Lubrication Council
Plans of.... 814
Industrial Management
Pequot mills.... 73
Industrial Plants
Visits to (Ed).... 206
Industrial Recovery
Fundamentals for.... 15
Neglected elements of.... 345
Restrictions to (Ed).... 274
Industrial Relations
Personality in.... 277
Industrial Relations Conference
Announcement of.... 464
Industrial Training
Cooperative system.... 418
Hobart Manufacturing Co. (S).... 448
Industry
Disease prevention in....90, 92, 95, 230
Mechanization in.... 303
Share of labor in (S).... 447
Waste in....561, 563
Inertia
Moment of, for pipes (S).... 309
Ingraham, Olin
Internal debt and liquid claims to wealth (BR).... 582
Insulating Materials. *See* Heat Insulating Materials
Interferometers
Precision measurements by.... 772
Internal-Combustion Engines. *See also* Airplane Engines; Automobile Engines
Axial (S).... 515
Combustion chamber (S).... 381
Combustion in (S).... 437
Detonation (S).... 310
Efficiencies (S).... 190
Fuels. *See* Engine Fuels
Improvements in (S).... 311
Supercharging (S).... 437
Thomas (S).... 660
International Electrochemical Commission
1935 meetings of.... 650

J

Jacobsen, S. Clark
Biography of.... 333
Discussion on paper by.... 388
Solid smokeless fuel.... 305
Jacobus, David S.
A.S.M.E. participation in forming steamship boiler rules (C).... 592
Honorary membership conferred upon.... 35
Portrait of.... 35
Jennison, Ernest N.
Biography of.... 395
McDonald observatory dome.... 349
John Fritz Medal
Award to Durand.... 814
Johnston, John
Applications of science to making and finishing of steel.... 79
Biography of.... 134
Portrait of.... 37
Thurston lecture by.... 79
Thurston lecture, presentation of.... 37
Jones, F. Robertson
Biography of.... 134
Occupational diseases.... 90
Journal of Applied Mechanics
Papers in.... 272
Junior Engineers
Relationship of older engineers to (Ed).. 537
Juran, J. M.
Biography of.... 670
Inspectors' errors in quality control.... 643

K

Kaltenborn, H. V.
Annual dinner speech by.... 36
Karapetoff, Vladimir
Wine makers and bottle makers (C).... 124
Karelitz, George B.
Biography of.... 332
Side oil flow in 120-degree bearing.... 292
Keith, Leigh S.
Elected secretary of Western Society of Engineers.... 672
Keller, H. H.
Recent trends in housing (abstract).... 573
Kerr, S. Logan
Turbines for Wheeler Dam (C).... 728
Keyes, Frederick G.
Equation of state for superheated steam. 164
Kimball, Dexter S.
Biography of.... 66
Portrait of.... 7
Prophets and panaceas.... 5
Towne lecture by.... 5
Klemin, A.
N.A.C.A. Langley Field conference.... 428
Kline, J. E.
Biography of.... 808
Desired characters of surface finishes.... 749
Kolb, J. F.
Apprentice training methods.... 421
Biography of.... 459
Kreitner, Hans
Water-hammer diagram.... 781

L

Labor
Hours of (Ed).... 275
Lamme Medal
Award to Wickenden.... 534
Langley Field
Conference, report of.... 428
Langmuir, Irving
Address by (Ed).... 467
Biography of.... 527
Mechanical properties of matter.... 486
Portrait of.... 510
Lawrence, T. E.
Epitaph of (Ed).... 401
Lehigh Valley Engineers' Club
Fifteenth anniversary meeting of, plans for.... 812
Lehman, Werner
Classification of lubricants (C).... 732
Leipzig Trade Fair
Announcement of.... 66
Letters and Comment....55, 122, 191, 257, 321, 388, 449, 592, 667, 726, 802
Lewis Institute
Director of retires.... 272
Licenses
Electronic, granting of (S).... 448
Lickert, Rensis
Suggestion systems (C).... 322
Lindemann, W. C.
Biography of.... 528
Photograph of.... 529
Linoleum Industry
Founder of.... 297
Liphshitz, I. N.
Biography of.... 204
Discussion on paper by (C).... 449
Sociology of invention (BR).... 734
Study of technical history.... 143
Locomotives
German streamlined (S).... 799
Three-cylinder, balancing of (S).... 723
Locomotives, Diesel. *See* Diesel Locomotives
Locomotives, Steam. *See* Steam Locomotives
Lubricants
Automotive (S).... 384
Lubricating Oil
Engines, rating quality of (S).... 661
Reclamation (S).... 115
Synthetic, from condensation products of ethylene (S).... 382
Lubrication
Aircraft engines (S).... 252
Bearings, 120-deg, side oil flow.... 292
Colloidal-graphite (S).... 184
Crankcase (S).... 381
Methods at Eastman Kodak plants.... 565
Research in (S).... 439
Steel mills.... 415
Steel plants.... 25
Turbine (S).... 316

LUBRICATION ENGINEERING COMMITTEE
Transferred to machine shop practice division........ 132

M

MACGREGOR, CHARLES W.
Biography of........ 265
Deflection of a long helical gear tooth.... 225
MACHINE DRIVE
Pivoted-motor........ 287
MACHINE PARTS
Failures in, stress analysis of........ 157
Screws (S)........ 586
MACHINERY
Brass mill (S)........ 188
Inevitableness of (Ed)........ 139
Stress analysis of failures in........ 157
Survey of potential markets (Ed)........ 274
MACHINE SHOP EQUIPMENT
Coining power press (S)........ 116
MACHINE SHOP PRACTICE
Discussion on (C)........ 124
Drilling copper tube plates (S)........ 662
Electroplated surfaces (S)........ 247
Fiber machinery (S)........ 48
Hard-facing with boron (S)........ 719
Reconditioning defective metal parts (S). 439
Surface finishing........(S) 312, 749
MACHINE TOOL CONGRESS, THIRD NATIONAL
Announcement of........ 464
MACHINE-TOOL INDUSTRY
Salute to (Ed)........ 610
MACHINE-TOOL SHOW
Cleveland, September, announcement of 604
Cleveland, September, report of........ 653
MACHINISTS
Training in Germany (S)........ 725
MAGNETIC TESTING
Airplane engine parts (S)........ 446
MAIN AWARD. *See* CHARLES T. MAIN AWARD
MANNERS
Men and (Ed)........ 205
MANOMETERS
Oil, for steam pressure measurement..... 633
MANUFACTURING
Dynamic production (C)........ 56
Economic lot sizes in (C)........ 55
MARBLETTE
Properties of........ 767
MARINE ENGINES. *See also* STEAM ENGINES, MARINE
Fuel consumption and maintenance costs of (S)........ 720
Stephen trials of (S)........ 256
MARSH, T. A.
Domestic heating (C)........ 259
MARTELLOTTI, M.
Biography of........ 527
Metal cutting........ 487
MARXISM
Foundations of (BR)........ 434
MASSACHUSETTS INSTITUTE OF TECHNOLOGY
Cooperative mechanical engineering course at........ 466
MASS PRODUCTION
Dimensional control........ 772
MATHEMATICS
Bessel functions (S)........ 49
MATHEWSON, J. S.
Prefabricated system, a new departure in all-wood housing (abstract)........ 578
MATSCHOSS, CONRAD
Resignation as director, V.D.I........ 336
MATTER
Mechanical properties of........ 486
MCCARTHY, R. H.
Economic lot sizes in manufacturing (C). 55
MCCLELLAN, WILLIAM
Biography of........204, 265
Commercial aspect of engineering........ 174
Selling one's services........ 237
MCELWEE, R. G.
Biography of........ 459
Discussion of paper by (C)........ 595
Modern cast iron........429, (C) 596
MCGREGOR, A. G.
Correct economy for machine age ....(BR) 235, (C) 451
MEASUREMENTS
Decimal dimensioning (S)........ 447
MEASURING INSTRUMENTS
Interferometers........ 772
MECHANICAL CATALOG
Publication of........ 746
MECHANICAL ENGINEERING
New curriculum in........ 483
MECHANICAL ENGINEERS' DAY, SAN DIEGO
Announcement of........ 461
MECHANISMS
Dimensional control........ 772
MEDICAL RESEARCH
Engineering methods in........ ....(Ed) 205, 217, (Ed) 402
METAL CUTTING
Bibliography on published........ 672
Handbook on........198, 536
METALLURGISTS
Engineers and........ 216
METALLURGY
Oxygenated air in (S)........ 587
METALS
Cracking of, process for testing resistance to (S)........ 589
Oil absorption by........ 413
Physical properties at low temperatures (S)........ 248
Properties at elevated temperatures (S).. 45
Protective coatings........ 355
Reconditioning (S)........ 439
Steels for steam engineering (S)........ 309
METALS CORROSION
Aluminum (S)........ 189
Prevention........ 355
Research on (S)........ 658
METALS CUTTING
Built-up edge, formation and function of 487
Structural steel........ 423
METAL SPRAYING
Molten (S)........ 189
METALS TESTING
Ductility of weld metal (S)........ 446
METALLURGY
Steel manufacture........ 79
METCALF, HENRY C.
Suggestion systems (C)........ 321
MEYER, ADOLPHE
Biography of........ 527
Portrait of........ 511
Rice lecture by (Ed)........ 467
Velox steam generator........ 469
MICROSTRUCTURE
Steel........ 625
MIDWEST ENGINEERING AND POWER EXPOSITION
Announcement of........ 400
MINE VENTILATION EQUIPMENT
Dust filters (S)........ 723
MOLLIER DIAGRAM
Bleeder turbine (S)........ 721
MOLLIER, RICHARD
Tribute to (Ed)........ 273
MONEL METAL
"K" alloy (S)........ 659
MOODY, LEWIS F.
Turbines for Wheeler Dam (C)........ 730
MOORHEAD, D. G.
Biography of........ 332
Dnieprostroy turbines........ 281
MORELAND, EDWARD L.
Head of Electrical Department at M.I.T. 396
"MORRO CASTLE"
Lessons of fire in........ 31
Thermostatic detectors (C)........ 191
MOSES, ADOLPH
Dynamical production analysis (C)...... 56
MOSS, SANFORD A.
Wine makers and bottle makers (C)..... 124
MOTOR BUSES
Diesel, electrical power transmission in (S) 520
MOTOR FUELS. *See* AUTOMOTIVE FUELS
MOTORS
Pivoted........ 287
MUNICH TECHNICAL UNIVERSITY
Hydraulic reports, of published in English 812
MUNITIONS. *See* GUNS
MURPHY, GEORGE M.
Biography of........ 204
Heavy hydrogen and heavy water....... 141

N

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS
Langley Field conference, report of...... 428
NATIONAL CASH REGISTER COMPANY
Industrial waste prevention........ 563
NATIONAL DEFENSE
Air warfare (S)........ 518
NATIONAL ENGINEERING SOCIETIES
Joint meeting of officers, announcement of 335
Joint meeting of officers, report on...... 459
Wilkes-Barre joint meeting of, announcement........ 608
NATIONAL INDUSTRIAL RECOVERY ACT. *See also* CAPITAL-GOODS INDUSTRIES; DURABLE-GOODS INDUSTRIES
Reports on restrictions of....(Ed) 2, (Ed) 3
NATURAL-GAS
Steam turbine use (S)........ 382
NRA CODES
Production restrictions in........ 9
NAUMBERG, ROBERT E.
John Price Wetherell Medal awarded to 395
NELSON, J. F.
Biography of........ 66
Code restrictions on machinery and production........ 9
NICHOLS, J. T.
Biography of........ 670
Metallic heat insulation........ 621
NICHOLS, W. W.
Portrait of........ 812
Testimonial to........ 812
NITRIC ACID
Formation in combustion of fuel oils (S) 661
NITRIDATION
Steel (S)........ 513
NOISE ABATEMENT
Engineers' interest in..(Ed) 538, (C) 732
NORRIS, CHARLES B.
New methods of construction for prefabricated houses (abstract)........ 576
NORTON, PAUL T., JR.
Economic lot sizes in manufacturing (C) 55
NYMAN, RICHMOND C.
Biography of........ 134
Labor-management cooperation in methods development........ 73

O

OBITUARIES. *See also* A.S.M.E., Members, Deaths of
Atterbury, William Wallace........ 789
Bond, George M. (Ed)........ 71
Parsons, Henry deB. (Ed)........ 138
Ricketts, Palmer C........(Ed) 3, 177
Sprague, Frank J. (Ed)........ 3
O'BRIEN, E. W.
Visits to Local Sections........ 202
OBSERVATORIES
Domes for........ 349
McDonald........ 349
OIL
Absorption by metals........ 413
Distillation from coal (S)........ 113
Fuel. *See* FUEL OILS
Lubricating. *See* LUBRICATING OIL
OIL AND GAS POWER
Progress in, 1934........ 99
OIL BURNERS
Domestic heating by........221, (C) 453
OIL-ENGINE POWER
1934 cost report available........ 672
OIL MANOMETERS. *See* MANOMETERS, OIL
OILS, FUEL
Cracked-residue, burning of........ 241
ORIFICES
Rounded-approach, use at low Reynolds' numbers........ 760
OSBORNE, N. S.
Steam research at the Bureau of Standards........ 162
OSCILLOGRAMS
Frequency measurement........ 215
OWENS, T. H.
Biography of........ 603
Salvage and reclamation........ 561
OXYGEN
Low-cost production of (S)........ 586

P

PALERMO, A. J.
Twenty-hour soap film........ 376

PALMER, VIRGIL M.
Biography of ... 603
Discussion of paper by ... (C) 122, (C) 192, (C) 321, (C) 732, (C) 802
Lubrication program ... 565

PARIS AIRCRAFT SHOW
Exhibits at (S) ... 436

PARISH, WILLIAM F.
Lubrication program (C) ... 802

PARKER, JAMES W.
Biography of ... 530
Photograph of ... 529

PARMELEE, JULIUS H.
Road and rail in forty countries (BR) ... 454

PARSONS, CHARLES A.
Tribute to (Ed) ... 273

PARSONS, HENRY DEB.
Tribute to (Ed) ... 138

PARTRIDGE, EVERETT P.
Biography of ... 333
Boiler-steel embrittlement ... 294

PASSAMAQUODDY POWER PROJECT
Description of ... 580

PATTERSON, WILLIAM F.
Biography of ... 459
Discussion of paper by (C) ... 597
National apprenticeship under way ... 420

PELLY, JOHN F.
Biography of ... 68
Discussion of paper by (C) ... 594
Steel-plant lubrication ... 25

PEQUOT MILLS
Cooperative labor-saving plan at ... 73

PERRY, THOMAS D.
Plywood and prefabricated plywood house units (abstract) ... 574

PERSONALITY
Business relations and ... 277

PETROGAS
Refrigerating qualities (S) ... 185

PHOTOELASTIC MATERIALS
Marblette ... 767

PIPE
Moment of inertia (S) ... 309
Welds, annealing by induction (S) ... 662

PIPE JOINTS
Refrigerating plant, Basle ice rink (S) ... 117

PIPES
Underground bituminous coatings for (S) 117

PIPING
Corrugated and creased-bend, strength and flexibility of (S) ... 796

PLASTICS
Directory of (S) ... 513

POTTER, A. A.
Portrait of ... 42
Research (BR) ... 600

POROSITY
Steel castings (S) ... 518

POWDERED COAL. *See* PULVERIZED COAL

POWER
Progress in, 1934 ... 99
Steam vs. hydro ... 403

POWER-PLANT ENGINEERING
Changes in existing plants..(S) 443, 444
Equalization of power and heat by steam compression (S) ... 587

POWER PLANTS
Colgate-Palmolive-Peet, changes in (S) .. 443
Connors Creek Station, revamping of (S) 444
Heat consumption vs. condensing-water temperature (S) ... 316
Heat losses in (S) ... 663
Hydraulic and heat combined (S) ... 43
Peak-load problem in (S) ... 252
Removal of sulphur from chimney gases (S) ... 519
Rowan Gotaverken exhaust turbine system for (S) ... 253
Schmidt propeller suction-draft system for (S) ... 252
Steam storage in (S) ... 797
Texas, University of (S) ... 185

POWER TRANSMISSION
Voith turbo (S) ... 516

PREFABRICATED HOUSING
Abstracts of papers on ... 571

PRESSES
Cold coining (S) ... 116
Forging, solid-frame (S) ... 520

PRINTING INDUSTRIES CONFERENCE
Proceedings of available ... 606

PRODUCER GAS. *See also* AUTOMOTIVE FUELS

PRODUCTION CONTROL
America's income and ... 507
Analysis of (C) ... 56

PROFESSIONAL SERVICES
Advertising for (Ed) ... 401

PROFESSIONAL SOCIETIES
Organization of (Ed) ... 401

PROFILOGRAMS
Surface finishes (S) ... 312

PROFIT
No substitute for (Ed) ... 342

PROPELLERS. *See* SHIP PROPELLERS

PROTECTIVE COATINGS
Molten-metal (S) ... 189

PROVIDENCE ENGINEERING SOCIETY
New home dedicated ... 64

PULVERIZED COAL
Bulk delivery (S) ... 381
Combustion zone (S) ... 793

PUMPS
Diesel-driven (S) ... 445
Water-jet (S) ... 114
Water-jet air, tests of (S) ... 588

PUMPS, CENTRIFUGAL
Casing design (S) ... 114

PUMPS, PROPELLER
Design (S) ... 114

PUMPS, ROTARY
High-vacuum (S) ... 53

PURDUE UNIVERSITY
Curricula at ... 398

PWA
Interest rates of ... 674

Q

QUALITY CONTROL. *See also* MASS PRODUCTION, Dimensional Control
Inspectors' errors in ... 643
Sampling inspection plans for ... 645
Telephone equipment ... 638
Textile mills ... 636

R

RADIATION. *See* HEAT LOSSES

RAILROAD LOCOMOTIVES. *See* LOCOMOTIVES

RAILROAD TRAINS
Burlington Zephyr (S) ... 445
"Comet" high-speed ... 479, 553
Diesel engines for ... 547
Light weight (S) ... 445
Superspeed (S) ... 519

RAUTENSTRAUCH, WALTER
Flexible budget (BR) ... 389

RAY, DAVID H.
The Master Science (C) ... 388

RAYON MANUFACTURE. *See* TEXTILE INDUSTRY

REEMPLOYMENT
A.S.M.E. program for ... 129

REES, ROBERT I.
Art of leadership (BR) ... 262

REESE, L. V.
Assistant secretary of A.E.C. ... 338

REFRACTORIES
Gas-retort (S) ... 247

REFRIGERANTS
Liquid, specific heats of (S) ... 317

REFRIGERATING MACHINERY
Rotary compressors for (S) ... 377

REFRIGERATORS
Electric, efficiency of (S) ... 799

REGISTRATION LAWS
Engineers ... 783

REGULA, ALBERT S.
Suggestion systems (C) ... 123

RESEARCH
Automotive fuels and lubricants (S) ... 384
Cavitation ... 211
Cinematography in (Ed) ... 341
Fuel engineering (S) ... 181
Lubrication (S) ... 439
Steam, at Bureau of Standards ... 162
Steam, at Massachusetts Institute of Technology ... 164

RESWICK, MAURICE
Lubrication program (C) ... 802

RICE, CALVIN W.
Council tribute to ... 37
Memorial lecture ... 36
Meyer lecture on (Ed) ... 467
Second memorial lecture ... 469

RICHARDS, C. R.
Elected president-emeritus of Lehigh University ... 676

RICHARDSON, CHAS. G.
Venturi and weir measurements (C) ... 667

RICHARDSON, MARION B.
"Analysis of Occupations of College Grade Compared to the Engineering Profession," mimeographed copies available ... 400

RICKETTS, PALMER C.
Obituary of ... 177
Portrait of ... 177
Tribute to (Ed) ... 3

RIEGEL, R. M.
Turbines for Wheeler Dam (C) ... 726

RIFLES
Firing phenomena, study of (S) ... 379

RIVETED JOINTS
Steel structures, tests of (S) ... 516

ROE, JOSEPH W.
John Hays Hammond (BR) ... 599

ROGERS, F. H.
Biography of ... 527
Discussion of paper by (C) ... 726
45,000-hp propeller turbine for Wheeler Dam ... 499

ROGERS, WILL
Spirit of St. Louis Medal to be awarded to ... 671

ROLLING MILL PRACTICE
Sheet and strip, 1934 developments in (S) 247

ROLLING MILLS
Brass, Canada (S) ... 188

ROSE POLYTECHNIC INSTITUTE
Fiftieth anniversary address by Dr. Hood 685

ROTARY COMPRESSORS
Refrigerating machinery (S) ... 377
Surth single-stage (S) ... 377

ROTORS. *See* ELECTRIC ROTORS

ROYAL SOCIETY OF ARTS
Prizes for inventions and essays ... 334

RUBBER TIRES
Stresses in ... 149

RUDDERS
Hydraulically operated (S) ... 794

S

SACKETT, R. L.
Biography of ... 528
Engineering education (BR) ... 262
Photograph of ... 529

SALISBURY, JOHN Q.
Biography of ... 603
Prevention of waste and reclamation of materials ... 563

SATURATED STEAM
Properties of (table) ... 163

SAYERS, ROYD RAY
Biography of ... 265
Prevention of occupational diseases ... 230

SCHNEIDER, HERMAN
Backgrounds of cooperative system ... 418
Biography of ... 459

SCHRECK, H.
Diesel engines (BR) ... 805

SCHROEDER, W. C.
Biography of ... 333
Boiler-steel embrittlement ... 294

SCIENCE
Application to steel manufacture ... 79

SCIENCE ADVISORY BOARD
Possibilities of (Ed) ... 69

SCOTT, C. F.
Elected chairman of E.C.P.D. ... 742

SCREWS
Hollow set and socket-head (S) ... 586

SEABURY, GEORGE T.
Portrait of ... 130

SECURITIES AND EXCHANGE COMMISSION
Engineer and (Ed) ... 71

SEELEY, LAUREN E.
Biography of ... 265
Discussion of paper by (C) ... 453
Oil burners and boilers ... 221

SEWARD, HERBERT L.
Biography of ... 68
"Morro Castle" disaster ... 31

SHARP, R. E. B.
Biography of ... 527
Discussion of paper by (C) ... 726
45,000-hp propeller turbine for Wheeler Dam ... 499

SHEWHART, W. A.
Probability and the physical world (BR) 327
SHIPPING, ENGINEERING, AND MACHINERY EXHIBITION
Announcement of 464
SHIP PROPELLERS
Garratt auxiliary drive for (S) 797
Melcher (S) 49
SHIPS
Fire prevention on 31
SHIRE, A. C.
Federal housing administration and recent developments in dwelling construction (abstract) 573
SHOUDY, W. A.
Biography of 527, 528
New curriculum in mechanical engineering 483
Photograph of 529
SILK MANUFACTURE. *See* TEXTILE INDUSTRY
SLAG
Boiler furnace, tapping of 367
SLIPHER, D. C.
Mechanical units for houses (abstract) 578
SMITH, E. DILLON
Control of distribution (C) 257
Corporate profits (BR) 601
Discussion of paper by (C) 56
Industrial profits (BR) 195
Statistics (BR) 456
SMITH, ED S., JR.
Venturi and weir measurements (C) 667
SMITH, HORACE F.
Biography of 603
Discussion of paper by (C) 732, (C) 802
Lubrication program 565
SMITH, J. F. DOWNIE
Biography of 808
Rounded-approach orifices 760
SMITH, LEIGHTON B.
Equation of state for superheated steam 164
SOAP FILM
Twenty-hour 376
SOCIAL SECURITY
Problem of 20
SOCIETY FOR THE PROMOTION OF ENGINEERING EDUCATION
Meeting announcement 336
New officers of 532
SOCIETY OF RHEOLOGY
New York meeting of, announced 676
SODERBERG, C. RICHARD
Biography of 204
Recent developments in steam turbines 165
SOLAKIAN, ARSHAG G.
Biography of 808
New photoelastic material 767
SOLDERS
Silver (S) 313
SOUDER, WILMER
Identification of firearms (BR) 804
SPECIAL LIBRARIES ASSOCIATION
Meeting announcement 336
SPECIFIC HEAT
Gaseous (S) 190
Water (S) 54
SPEED REDUCERS. *See also* GEARS, Worm Reduction
A.G.M.A. adopts rating of 674
Planetary (S) 382
SPELLER, FRANK N.
Biography of 395
Influence of protective layers on life of metals 355
SPICER, C. W.
Honorary D.S. degree conferred upon 676
SPIRIT OF ST. LOUIS MEDAL
Award to Rogers 671
SPRAGUE, FRANK J.
Tribute to (Ed) 3
STAINLESS STEEL
Silver-soldered (S) 659
STARKWEATHER, J. D.
Biography of 670
Oil manometer 633
STEAM
Electric generation of (S) 798
Research, at Bureau of Standards 162
Research, at Massachusetts Institute of Technology 164
STEAM ACCUMULATORS
Lagged, heat losses in (S) 663
STEAM BOILERS
Producer-furnace-fired Lancashire (S) 722
Vertical, protection of smoke passages of (S) 798
Wooden 148
STEAM CONDENSERS
Pressure measurement in 633
STEAM ENGINES
Sonnleithner single-cylinder reciprocating (S) 725
Sulzer back-pressure (S) 724
STEAM ENGINES, MARINE
Elsinore (S) 318
STEAM GENERATORS. *See* BOILERS; STEAM, Electric Generation of
STEAM LOCOMOTIVES
Building industry use (S) 254
STEAM PIPE
Corrugated and creased-bend (S) 796
STEAM POWER
Hydroelectric vs. 403
Progress in, 1934 99
STEAM POWER PLANTS
District heating, Zurich (S) 117
STEAMSHIPS
Boiler location (S) 441
STEAM TABLES
Third international conference on 710
STEAM TURBINES
Back-pressure, tests on (S) 384
Blades (S) 386
Bleeder (S) 721
Bleeder, graphic presentation of steam consumption of (S) 799
Bleeder, tests on (S) 384
Compressed-air (S) 317
Developments in 165
High-pressure, tests on (S) 663
Huttner (S) 120
Lubrication (S) 119
Natural gas in (S) 382
Oil in (S) 119
Regulation (S) 185
Simplex unit marine geared (S) 184
Testing 705
Testing, hydraulic analogies in (S) 110
STEAM TURBINES, MARINE
Simplex geared unit (S) 184
STEEL. *See also* ALLOY STEEL
Creep of, as influenced by microstructure 625
Cutting of 423
Embrittlement 294
Heat-treated, effects of grooves and fillet shapes on endurance of (S) 116
Nitriding of (S) 513
Notch sensitiveness of (S) 309
Steam engineering (S) 309
STEEL CASTINGS
Porosity (S) 518
STEELE, SYDNEY
Biography of 738, 808
Rounded-approach orifices 760
Separation of radiation and convection heat losses 700
STEEL MANUFACTURE
Applications of science to 79
STEEL MILLS
Lubrication 415
Velox boilers in (S) 254
STEEL PLANTS
Lubrication 25
STEEL STRUCTURES
Riveted joints, tests of (S) 516
STEVENS ENGINEERING CAMP
Economic conference for engineers, August 10–18, announcement of 536
STEVENS INSTITUTE
Gantt exercises at 39
STIMSON, H. F.
Steam research at the Bureau of Standards 162
STOKERS
Mechanical automatic operation of (S) 383
Underfeed, Great Britain 695
Underfeed, water-cooled 761
STRAIGHT, D. ARTHUR
Suggestion systems (C) 192
STRASSBURGER, J. H.
Biography of 395
Discussion of paper by (C) 667
Slag tapping of flat-bottom boiler furnaces 367
STRATTON, REUEL C.
Biography of 134
Toxic dusts 95
STRENGTH OF MATERIALS
Estimating ultimate (S) 584
STRESSES
Machine parts 157
Overstrained materials (S) 791
Rubber tires 149
Shearing, soap-film studies of (S) 513
Steel (S) 309
"STRETCHOUT" SYSTEMS
Pequot mills 73
STROBOSCOPE
Study of flow of air by 228
Study of life by (Ed) 205
SUGGESTION SYSTEMS
Discussion on (C) 122, (C) 192, (C) 321
SUMMER INSTITUTE FOR SOCIAL PROGRESS
Lectures at 396
SUPERCHARGING
Wibu method of (S) 437
SUPERHEATED STEAM
Equation of state for 164
SUPERHEATERS
Houlet annular (S) 384
Manufacture at Swindon Works (S) 384
SURFACE FINISHES
Desired characters of 749
SWASEY, AMBROSE
Washington Award, 1935, awarded to 202

T

TATNALL, ROBERT R.
Biography of 332
Discussion of paper by (C) 593
Pivoted-motor drive 287
TAYLOR, C. FAYETTE
Internal-combustion engine (BR) 262
TAYLOR, FRANK A.
Wooden steam boilers 148
TECHNICAL LITERATURE
Improvements in (Ed) 609
TECHNOLOGICAL PROGRESS
Budgeting effort toward (Ed) 69
TELEPHONE EQUIPMENT
Quality of control of 638
TERRY, R. V.
Biography of 332
Dnieprostroy turbines 281
TEXTILE FIBERS
Vegetable, effect of cation-active fats on (S) 447
TEXTILE INDUSTRY
Work assignment in (Ed) 677, 679
TEXTILE MACHINERY
Casablancas' compound drafting machine (S) 53
Cinematographic research (S) 446
TEXTILE MILLS
Quality control in 636
Waste disposal 631
TEXTILE WORK ASSIGNMENT BOARDS
Reports of (Ed) 468
THERMODYNAMICS
Compressed liquid water 618
Ejector cycle (S) 121
Heat transmission in finned cylinders, effect of baffles on (S) 591
Name for "heat content" 270
THOMSON, ELIHU
Honored by V.D.I. 336
THRESHER, B. A.
Mechanization in industry 303
THURSTON, ROBERT HENRY
Sixth lecture on relation between engineering and science 37, 79
THYRATRON MOTOR
Features of (S) 45
TIDAL POWER
Passamaquoddy project 580
TIMOSHENKO, S.
Vibrations in bridges (BR) 58
TIRES
Rubber. *See* RUBBER TIRES
TOBIN, THEO. A.
"General Impressions of a Junior Engineer" (Ed) 537
TOLERANCES
Principles of 774
TOWNE, HENRY ROBINSON
Seventh lecture on engineering and economics 5
TRAINS. *See* RAILROAD TRAINS
TRANSMISSIONS
Voith turbo (S) 516
TRAVER, G. W.
Prefabricated system, a new departure in all-wood housing (abstract) 578
TRINKS, W.
Heat calculations for furnaces (BR) 735
TRUCKS
Diesel-driven, Germany (S) 518
TUCKER, DONALD S.
America's output and income (BR) 507

TUCKER, ROSS F.
Financial aspects of housing (abstract)... 572
TUNGUM ALLOY
Composition and properties of (S)....... 658
TURBINE BLADES. *See* STEAM TURBINES, BLADES
TURBINES. *See also* STEAM TURBINES
Boulder Dam.......................... 539
Dnieprostroy.......................... 281
Hydraulic. *See* HYDRAULIC TURBINES
Industrial, developments in (S)......... 313
Internal-combustion (S)................ 311
Lubrication (S)....................... 316
Propeller, Wheeler Dam................ 499
Steam. *See* STEAM TURBINES
Steam, model, tests of (S).............. 110
TURBINES, HYDRAULIC
Comparison of (table)................. 539
TURBINES, STEAM. *See* STEAM TURBINES
TURBULENCE
Study of.............................. 407
Wind tunnels (S)...................... 442

U

UNFIRED PRESSURE VESSELS
A.P.I.-A.S.M.E. code for, relation to boiler construction code.............. 131
UNITED ENGINEERING TRUSTEES, INC.
Officers of elected..................... 200
U.S.S.R.
Dnieprostroy development............. 281
UNITS. *See* ELECTRIC UNITS
UNIVERSAL JOINTS
Mollart (S)........................... 796
UNWIN, WILLIAM CAWTHORNE
Memorial funds solicited............... 811
Portrait of............................ 811
UPHAM, E. W.
Modern cast iron (C).................. 595

V

VAN HENGEL, G. H.
Biography of.......................... 670
Oil manometer........................ 633
VAN TONGEREN, HERMAN
Biography of.......................... 808
Modern dust collector.................. 753
VAPORS
Physical and toxic properties of common (table)............................. 232
VENTURI METERS
Alden hydraulic laboratory............. 369
VON KARMAN, TH.
Biography of.......................... 459
Some aspects of turbulence problem..... 407
VOORHEES, S. F.
Election as president of the American Institute of Architects............... 461

W

WAGES
Prices and............................ 346
WALTON, FREDERICK
Biography of.......................... 297
Portrait of........................... 297
WAR
Weapons of (Ed)...................... 342
WARFARE
Air (S)............................... 518
WARNER MEDAL. *See* WORCESTER REED WARNER MEDAL
WASHINGTON AWARD
Awarded to Ambrose Swasey........... 202
WASTE
Industrial.........................561, 563
WASTE HEAT
Furnace-gas (S)...................... 383
WATER
Heavy................................ 141
Specific heat of (S)................... 54
WATER HAMMER
Kreitner's diagram for problems of...... 781
WATER HEATERS
Heat transmission in (S)............... 319
WATER PIPE LINES. *See* CONDUITS
WATER POLLUTION
Textile wastes and.................... 631
WATER-TUBE BOILERS. *See* BOILERS, WATER-TUBE
WEAVING
Silk and rayon, elementary operations in 684
WEIR MEASUREMENT
Alden hydraulic laboratory............. 369
WEISSELBERG, ARNOLD
Jobs for young engineers (C).......... 258
WELDING
Bronze to steel (S).................... 796
Oxy-ferrolene (S)..................... 386
Sheet (S)............................. 320
Survey of literature on................ 606
WELDING CONFERENCE
Lubbock, Texas, Feb. 14 and 15, announcement of...................... 132
WELDS
Pipe, annealing by induction (S)........ 662
WESTERN SOCIETY OF ENGINEERS
New secretary elected.................. 672
WESTINGHOUSE, GEORGE
Anecdotes about (Ed)................. 677
WETHERELL MEDAL
Award to Naumberg................... 395
"WHAT'S GOING ON"
Importance of (Ed).................... 137
WHEELER DAM
Turbine for........................... 499
WHITE, W. M.
Biography of.......................... 603
Construction of 115,000-hp Boulder Dam turbines............................ 539
WHITNEY, WILLIS R.
Edison medal awarded to.............. 132
WICKENDEN, WILLIAM E.
Engineering not a caste................ 433
Lamme medal awarded to.............. 534
WILKES-BARRE, PA.
Meeting and inspection trip, September 28, announcement of............... 608
WILLIAMS, C. C.
Elected president of Lehigh University.. 676
WILLIAMS, D. G.
Biography of.......................... 738
Effect of insulation.................... 687
WIND TUNNELS
Turbulence in (S)..................... 442
"WINE MAKERS"
Discussion on (C)..................... 191
Karapetoff parable of (C).............. 124
WOOLEN MILLS
Flue gas scrubbers for................. 361
WORCESTER REED WARNER MEDAL
Presentation of....................... 36
WORM GEARS
Manufacturing developments (S)........ 440
WYMAN, L. L.
Biography of.......................... 670
Creep of steels........................ 625

Y-Z

YELLOTT, JOHN I., JR.
Portrait of........................... 41
YOUNG, A. H.
Gantt medal awarded to............... 396
ZIMMERMAN, J. H.
Biography of.......................... 459
Cutting structural steel................ 423

Index to A.S.M.E. Transactions

Volume 57, 1935

The A.S.M.E. Transactions for 1935 was issued monthly, four of the twelve issues being the *Journal of Applied Mechanics*, the page numbers for which are preceded by the letter A.

Each paper in the other eight issues is given a letter symbol (appearing at the top of the page) which identifies the Division or committee sponsoring it. The key to these symbols follows: AER (Aeronautics), FSP (Fuels and Steam Power), HYD (Hydraulics), IS (Iron and Steel), MSP (Machine-Shop Practice), OGP (Oil and Gas Power), PRO (Process), RR (Railroad), PTC (Power Test Codes), and RP (Research). These symbols are accompanied in each case by the volume number of Transactions and the number of the paper; i.e., HYD-57-11 indicates the eleventh paper sponsored by the Hydraulic Division to be published in volume 57 of Transactions.

The Record and Index for the year 1935 was issued as supplements, one in February and one in August, 1935, and one in January, 1936. The page numbers for these supplements are designated by the symbol RI.

(D) as used in this index denotes a discussion of a paper.

Adams, Harold Francis
Obituary of....RI-42

Administrative Committee, Special, A.S.M.E.
Members of.... RI-8

Advertising Revenue Committee, A.S.M.E.
Members of.... RI-7

Advisory Board on Standards and Codes, A.S.M.E.
Members of.... RI-7
Report of....RI-107

Advisory Board on Technology, A.S.M.E.
Members of.... RI-7
Report of....RI-106

Aeronautical Division, A.S.M.E.
Committees of.... RI-9
Representatives on other activities.... RI-9,

Air Conditioning
Psychrographs....229, (D) 557

Aircraft
Fusion welding.... 389

Air Ducts
Fan inlet and discharge.... 429

Air Flow. *See* Flow of Air.

Airfoils
Boundary layers of.... A-21
Double-roof profile.... A-24
Lift coefficient (D).... 192
Maximum-lift coefficient of.... A-21
Single-roof profile.... A-23
Wind tunnel tests of.... A-26

Air Preheating
Stokers (D).... 351

Air-Water Mixtures
Chart of.... 45

Alfred Nobel Prize
A.S.M.E. representative on....RI-30
Report on....RI-115

Allan, George W.
Pitting due to rolling contact (D)....A-110

American Association for the Advancement of Science
Report of A.S.M.E. representatives on..RI-110

American Bureau of Welding
A.S.M.E. representatives on....RI-29
Report of A.S.M.E. representative on..RI-111

American Engineering Council
A.S.M.E. representatives on....RI-29
Report of A.S.M.E. representatives on.RI-111

American Society of Mechanical Engineers
Absorption of radiant heat in boiler furnaces, report of committee on.... 541
Assistant secretaries of.... RI-5
Awards, recipients of....RI-30
Awards, statements about....RI-30
Codes, published by, in 1935....RI-121
Committees, index to....RI-33
Committees, members of.... RI-5
Committees, reports of....RI-83
Committees, technical, list of publications of....RI-123
Council, members of.... RI-5
Council, report of....RI-81
Council, special committees of.... RI-7
Editor of.... RI-5
Honorary members of....RI-32
Indexes to publications....RI-121
Local Sections, executive committees of.RI-12
Medal, recipients of....RI-30
Medal, statement about....RI-30
Meetings, 1935, papers presented at....RI-121
Members, obituaries of....RI-37
Midwest and Mid-Continent offices of..RI-12
Nominating committee groups.... RI-8
Officers of.... RI-5
Office, service rendered by....RI-116
Papers presented at 1935 meetings of..RI-121
Past-presidents of....RI-32
President, annual address of....RI-77
President, biography of.... RI-3
President, photograph of.... RI-2

American Society of Mechanical Engineers (*continued*)
Professional Division Committees, members of.... RI-9
Publications, indexes to....RI-121
Representatives on other activities....RI-7, 9, 21, 25, 27, 28, 29
Representatives on other activities, reports of....RI-110
Special committees, members of.... RI-6
Standards, published by, in 1935....RI-121
Standing Committees, members of.... RI-5
Student Branches, list of....RI-18
Technical Committee publications, list of....RI-123
Technical Committees, Special, members of....RI-20
Transactions, depositories for....RI-117
Transactions, index to....RI-134

A.S.M.E. Medal
Recipients of....RI-30
Statement about....RI-30

American Standards Association
A.S.M.E. representatives on....RI-29
Report of A.S.M.E. representative on..RI-113

American Year Book Corporation
A.S.M.E. representative on....RI-29

Anemometers
Water velocity measurements with.... A-87

Applied Mechanics Division, A.S.M.E.
Committees of.... RI-9

Archers
Buckling of flat.... A-17

Armacost, W. H.
Experimental investigation of heat absorption in boiler furnaces.... 541

Armour, James W.
Economics of preheated air for stokers (D).... 351

Arnold, Norman
New basis for rating of roller-chain drives (D).... 443

Automatic Control
Principle factors of.... 159

Awards, A.S.M.E.
Recipients of....RI-30
Statements about....RI-30

Awards Committee, A.S.M.E.
Members of.... RI-5
Report of....RI-97

Awards, Joint
Reports on....RI-115

Axles
Fatigue of railway.... A-69
Fatigue tests, German State Railways.... A-1
Railroad cars, fatigue failure of, at wheel fits.... A-1

B

Babione, Marion Eugene
Obituary of....RI-42

Bachelder, Joseph E.
Obituary of....RI-42

Bailey, E. G.
Pulverized fuel-burning experiences at Buzzard Point Station (D).... 374

Bailey, N. P.
Discussion on paper by.... 188

Balancing
Reciprocating machines.... A-81

Balancing Machines
Dynamic (D).... 140

Barber, Everett M.
Current practice in pressures, speeds, clearances, and lubrication of oil-film bearings (D).... 358

Barnhart, A. C.
Discussion on paper by.... 190

Barre, Herbert A.
Obituary of....RI-42

Barrett, Charles S.
Fatigue of shafts at fitted members, with related photoelastic analysis (D).... A-69

Bars
Flat curved, buckling of.... A-17

Bartlett, G. M.
Discussion on paper by.... 443
New basis for rating of roller-chain drives. 97

Bassett, William Hastings
Obituary of....RI-43

Baudry, R.
Effects of side leakage in 120-degree centrally supported journal bearings (D). 136
Journal bearing performance....A-121

Baumgartner, Henry Howell
Obituary of....RI-43

Bean, H. S.
Air flow in fan-discharge ducts (D).... 347
V-notch weir for hot water (D).... 250

Bearing Metals
Performance of....35, (D) 380

Bearings
Film lubrication in.... A-59
Lubrication.... 169
Oil-film (D).... 355
Pitting in anti-friction.... A-49
Side leakage in (D).... 135
Sleeve, oil film in.... A-59

Bearings, Journal
Loading and friction of.... 169
Lubrication (D).... 135
Performance of....A-121

Bearings, Thrust
Loading and friction of.... 169

Beers, R. L.
Economics of preheated air for stokers (D).... 354

Bending. *See* Spring Wire.

Bennett, J. S.
Economics of preheated air for stokers (D) 353
Ten years of stoker development at Hudson Avenue (D).... 376

Berry, C. Harold
Steam-turbine leaving losses and vacuum corrections (D).... 555

Bibliographies
Automatic control.... 163
Bending and torsion fatigue tests.... 515
Ceramic industry.... 441
Film lubrication in sleeve bearings.... A-64
Flow of water....211, 226, 228, 316
Flue dust and cinders in chimneys.... 9
Fluid mechanics.... A-30
Generation of steam and electricity.... 17
Heat absorption in boiler furnaces.... 553
Heat transmission.... 470
Journal bearing performance....A-127
Journal lubrication.... 186
Lubrication....A-153
Pipe-bend stresses.... 401
Pipe-line stresses.... 168
Surface rolling....A-136
Thermodynamics.... A-66
Torsional stresses in shafts....A-155
Vibrations and sound....A-104
Viscosity of water and superheated steam 400

Biography Advisory Committee, A.S.M.E.
Members of.... RI-8

Biot, M. A.
Distributed gravity and temperature loading in two-dimensional elasticity replaced by boundary pressures and dislocations.... A-41

Blades. *See* Steam Turbines, Blades.

Blaine, Joseph Robert
Obituary of....RI-43

Blowers
Propeller, airfoil type.... 417

Board of Review (Delinquent Members), A.S.M.E.
Members of RI-7
Report of RI-107

Boiler Code Committee, A.S.M.E.
Members of RI-6
Report of RI-108

Boiler Code Conference Committee, A.S.M.E.
Members of RI-6

Boiler Code Executive Committee, A.S.M.E.
Members of RI-6

Boiler Code Subcommittee, A.S.M.E.
Members of RI-6

Boiler Codes
Published by A.S.M.E. in 1935 .... RI-121

Boiler Furnaces
Heat absorption in 517
Heat absorption in, computing 531
Heat absorption in, experimental investigation of 541
Radiation 105
Test results 541

Boilers
Forced-circulation, flow distribution in (D) 256
Hudson Avenue Station, tests of (D) 265

Boiler Tubes
Rolling-in of 145

Bollay, W.
Review of recent research work in fluid mechanics A-28

Bond Issue Committee, A.S.M.E.
Members of RI-7
Report of RI-108

Books Reviewed
Aerodynamic theory A-78, A-156
Analytical geometry of three dimensions A-158
Anecdotal history of the science of sound A-156
Atomic structure and spectral lines A-76
Conjugate functions for engineers A-119
Distortion of metal crystals A-115
Eine Losung für die Berechnung der biegsamen rechteckigen Platten (A General Solution for Rectangular Plates.) A-40
Fabricated welded steel construction A-75
Festigkeitslehre mittels spannungsoptik A-118
From Galileo to cosmic rays A-79
Fundamentals of hydro- and aeromechanics A-118
Hardness of metals and its measurement A-157
Higher mathematics for engineers and physicists A-159
Hydromechanik—Volume 2 .... A-40
International conference on physics A-115
Kristallplastizität mit besonderer berüchsichtigung der metalle A-115
Mechancial vibrations A-39
Metallic arc welding A-158
New pathways in science A-116
Noise. A comprehensive survey from every point of view A-158
Practical solution of torsional vibration problems A-119
Rayleigh's principle and its applications to engineering A-76
Resistance of materials A-158
Riveting and arc welding in ship construction A-75
Science and social needs A-159
Seaplane design A-77
Second report of the steel structures research committee A-75
Siphon spillways A-159
Statik und dynamik der schalen A-79
Story of energy A-120
Technische Thermodynamik A-38
Theory of elasticity A-38
Theory of functions as applied to engineering problems A-117
Thermodynamics of electrical phenomena in metals A-160
Through space and time A-80
Torsional vibration A-39

Boundary Layer, See Airfoils, Boundary layer of

Bowen, David Rees
Obituary of RI-44

Broaching
Surface (D) 258

Bromberg, Jacob
Discussion on paper by 239

Brooks, Henry K.
Obituary of RI-44

Brownlie, David
Test performance of Hudson Avenue's most recent steam-generating units (D) 269

Bryson, Tandy A.
Mathematical solution of rotor-balancing problem (D) 239

Buckland, B. O.
Discussion on paper by 251

Buckling. See also Bars, Flat curved, buckling of; Plates, Curved, buckling of
Cylindrical shells (D) 144

Buzzard Point Plant
Pulverized-fuel burning at (D) 374

C

Cain, B. S.
Safe operation of high-speed locomotives 471

Calderwood, James Park
Obituary of RI-44

Caldwell, W. E.
Discussion on paper by 265

Calorimeters
Radiation 105

Calvin W. Rice Memorial Committee, A.S.M.E.
Members of RI-8
Report of RI-110

Capital Goods Industries Committee, A.S.M.E.
Members of RI-7
Report of RI-109

Carbon Monoxide
High-purity generator 468

Carrier, W. H.
Air flow in fan-discharge ducts (D) 350

Carruthers, John L.
Drying problems of ceramic industry 439

Cather, J. H.
Cooperation between industrial and public-utility companies in generating steam and electricity (D) 17

Cavitation
Hydraulic turbines 455
Research on A-29
Research on, Massachusetts Institute of Technology 423
Research on, Princeton University 425

Centrifugal Fans. See Fans, Centrifugal

Ceramic Industry
Drying problems 439

Certificates of Indebtedness, Committee, A.S.M.E.
Members of RI-7

Chain Drives
Roller, rating of .... 97, (D) 443

Chamberlain, Adams Burton
Obituary of RI-45

Charles T. Main Award
Recipients of RI-31
Statement about RI-30

Chimneys
Calculation and dispersion of flue dust and cinders from 5

Christie, A. G.
Cooperation between industrial and public-utility companies in generating steam and electricity (D) 17
High-pressure steam and binary cycles as means of improving power-station efficiency (D) 141
Pulverized fuel-burning experiences at Buzzard Point Station (D) 375
Test performance of Hudson Avenue's most recent steam-generating units (D) 270

Church, Austin H.
Temperature history during constant-rate heating of solid circular cylinder A-96

Cinders
Dispersion from chimneys 5

Citizenship Manual Committee, A.S.M.E.
Members of RI-7
Report of RI-109

Clark, A. B.
High-pressure steam and binary cycles as means of improving power-station efficiency (D) 142

Clark, Walton
Obituary of RI-45

Coal
Grindability of (D) 190

Coal, Study of
A.S.M.E. representatives on RI-30

Coffin, William Jay
Obituary of RI-45

Cold-Working. See Steel, Cold-Working

Cole, C. S.
New method of investigating performance of bearing metals (D) 380

Cole, Edward S.
Pitot-tube practice 281

Colleges, Relations With, Committee, A.S.M.E.
Members of RI-5
Report of RI-96

Collett, Samuel Dunlap
Obituary of RI-45

Columns. See Cylindrical Shells

Committees, A.S.M.E.
Index to RI-33
Reports of RI-83

Committees, Professional Divisions, A.S.M.E.
Members of RI-9

Committees, Special, A.S.M.E.
Members of RI-6

Committees, Standing, A.S.M.E.
Members of RI-5

Connelly, John R.
Discussion on paper by 380
Effects of side leakage in 120-degree centrally supported journal bearings (D) 137
New method of investigating performance of bearing metals 35

Constitution and By-Laws Committee, A.S.M.E.
Members of RI-5
Report of RI-96

Control. See Automatic Control

Cooper, W. S.
Discussion on paper by 265
Fluid-meter nozzles (D) 254

Cooperative Relations Committee, A.S.M.E.
Members of RI-7
Report of RI-109

Cope, E. T.
Rolling-in of boiler tubes 145

Cornelius, Ernest Harry
Obituary of RI-45

Council, A.S.M.E.
Executive committee of RI-5
Members of RI-5
Report of RI-81
Special committees of RI-7

Craig, Ollison
Relative grindability of coal (D) 191

Cramer, Albert Nelson
Obituary of RI-46

Croft, Huber O.
Calculation of dispersion of flue dust and cinders from chimneys 5
Radiation intensities and heat transfer by radiation in boiler furnaces 105

Crooker, Ralph
Obituary of RI-46

Crystals
Plastic deformation of A-33

Current Meters
Behavior in flowing water 295
Hydroelectric power plants 303
Limitation of 203
Precise flow measurements with 227
Water measurement with 59

Cutler, Frank Garfield
Obituary of RI-46

Cycles
Binary (D) 141

Cylinders
Solid circular, heating of A-96
Strength under concentrated loading A-147

Cylindrical Shells
Buckling of (D) 144
Instability of (D) 259

D

Dahlstrom, F. P.
Current practice in pressures, speeds, clearances, and lubrication of oil-film bearings (D) 357

Dale, Orton Goodwin
Obituary of RI-47

Dall, Albert H.
Current practice in pressures, speeds, clearances, and lubrication of oil-film bearings (D) 355

Daniel Guggenheim Medal
Report on RI-115

Daniel Guggenheim Medal Fund, Inc.
A.S.M.E. representatives on RI-29

DAUGHERTY, ROBERT L.
Some physical properties of water and other fluids ... 193

DAVIDENKOFF, N. N.
Closure to discussion on paper by ... 134

DAVIS, L. M.
Cavitation testing of model hydraulic turbines and its bearing on design and operation ... 455

DAVISON, CYRUS EATON
Obituary of ... RI-47

DEAN, WILLIAM EDWARD, JR.
Obituary of ... RI-47

DEFORMATION
Tensile tests under high velocities of ... A-33

DEHAVEN, GEORGE PASS
Obituary of ... RI-47

DELINQUENT MEMBERS, BOARD OF REVIEW, A.S.M.E.
Members of ... RI-7
Report of ... RI-107

DEN HARTOG, J. P.
Hydromechanics (BR) ... A-40
On hydrodynamic analogy of torsion ... A-46
Statics and dynamics of shells (BR) ... A-79

DENT, JOHN A.
Steam-turbine leaving losses and vacuum corrections (D) ... 555

DESIGN DATA ... A-35, A-67, A-106, A-154

DICKINSON, E. D.
Fluid-meter nozzles (D) ... 255

DIESEL-ENGINE PLANTS
Power cost data for 1933 ... 69

DILLON, EDWARD H. J.
Obituary of ... RI-48

DILLON, R. E.
Discussion on paper by ... 351

DOEPKE, FRED
Obituary of ... RI-48

DONNELL, L. H.
Discussion on paper by ... 144

DRAFT GEARS. *See* RAILROAD ROLLING STOCK

DREWRY, M. K.
Pulverized fuel-burning experiences at Buzzard Point Station (D) ... 375

DREYFUS, E. D.
Cooperation between industrial and public-utility companies in generating steam and electricity (D) ... 18

DRISCOLL, JOHN M.
Discussion on paper by ... 376
Ten years of stoker development at Hudson Avenue ... 49

DRYING
Air, graphical analysis of ... 41
Classification of ... 41

DUNTEN, ROSCOE IRVING
Obituary of ... RI-48

DYCKERHOFF, A. H.
Cooperation between industrial and public-utility companies in generating steam and electricity (D) ... 19

E

EATON, GEORGE C.
Economics of preheated air for stokers (D) ... 353
Ten years of stoker development at Hudson Avenue (D) ... 378

EATON, H. N.
V-notch weir for hot water (D) ... 249

EBAUGH, N. C.
Discussion on paper by ... 374

ECONOMIC STATUS OF THE ENGINEER COMMITTEE, A.S.M.E.
Members of ... RI-7
Report of ... RI-109

EDUCATION AND TRAINING FOR THE INDUSTRIES COMMITTEE, A.S.M.E.
Members of ... RI-6
Report of ... RI-98

EGLI, ADOLF
Discussion on paper by ... 445
Leakage of steam through labyrinth seals ... 115

EKSERGIAN, R.
Discussion on paper by ... 128
Practical solution of torsional vibration problems (BR) ... A-119
Tractive effort of steam locomotives (locomotive ratios—II) (D) ... 364

ELASTICITY
Boussinesq formula ... A-69

ELASTICITY (*continued*)
Hydrodynamic analogy of torsion ... A-46
Steel at high temperatures ... 1
Two-dimensional ... A-41

ELASTIC-POINT THEORY
Application to pipe calculations ... 165

ELECTRICITY
Industrial and public-utility cooperation in generation of ... 11

ELLENWOOD, FRANK O.
High-pressure steam and binary cycles as means of improving power-station efficiency (D) ... 143
Test performance of Hudson Avenue's most recent steam-generating units (D) 270

EMPLOYMENT COMMITTEE, A.S.M.E.
Members of ... RI-7

ENDURANCE OF METALS
German tests of ... A-2
Westinghouse tests of ... A-3

ENGINEERING FOUNDATION
A.S.M.E. representatives on ... RI-29
Report of A.S.M.E. representatives on ... RI-113

ENGINEERING HISTORY COMMITTEE, A.S.M.E.
Members of, to be appointed ... RI-7
Report of ... RI-109

ENGINEERING REGISTRATION, NATIONAL BUREAU OF
A.S.M.E. representative on ... RI-29

ENGINEERING SOCIETIES EMPLOYMENT SERVICE
A.S.M.E. representatives on ... RI-29
Report of A.S.M.E. representative on ... RI-114

ENGINEERING SOCIETIES MONOGRAPHS COMMITTEE
A.S.M.E. representatives on ... RI-30

ENGINEERS' COUNCIL FOR PROFESSIONAL DEVELOPMENT
A.S.M.E. representatives on ... RI-29
Report of A.S.M.E. representatives on ... RI-114

ENGINEERS' NATIONAL RELIEF FUND
A.S.M.E. representative on ... RI-29
Report of A.S.M.E. representatives on ... RI-114

ENGLE, M. D.
Discussion on paper by ... 351

EVANS, THOMAS H.
Exact construction of ($\sigma_1 + \sigma_2$)-network from photoelastic observations (D) ... 139

EXTENSOMETERS
Lateral, for fatigue tests of metals ... A-9

F

FANS. *See also* BLOWERS
Axial-flow (D) ... 343
Axial-flow, high-pressure, design and performance of ... 383
Exhaust, testing (D) ... 374
Measurement of air-flow in inlet and discharge ducts ... 429
Propeller, design of ... 197
Quantity-head characteristic of ... 433

FANS, CENTRIFUGAL
Performance of (D) ... 244

FARKELL, GEORGE C.
Obituary of ... RI-48

FARMER, THOMAS
Obituary of ... RI-48

FATIGUE
Axles, railroad cars ... A-1
Railroad car axles ... A-1
Shafts at fitted members ... A-1

FATIGUE OF METALS. *See* METALS FATIGUE

FATIGUE TESTS. *See* SPRING WIRE

FEEDWATER TREATMENT
Prevention of turbine-blade deposits by ... 447

FINANCE COMMITTEE, A.S.M.E.
Members of ... RI-5
Report of ... RI-83

FILLETS
Stress concentration factors for ... A-67

FINLEY, JOHN H.
Calvin W. Rice's contribution to international friendliness ... RI-40

FISHER, EDWIN E. A.
Obituary of ... RI-49

FISHER, F. F.
Rolling-in of boiler tubes ... 145

FITTERER, J. C.
Design of light-weight trains (D) ... 129

FLAGG, STANLEY GRISWOLD
Obituary of ... RI-49

FLAMES
Nonluminous, radiation from ... 463

FLANDERS, RALPH E.
Biography of ... RI-3
New pioneers on a new frontier ... RI-77
Photograph of ... RI-2

FLEMING, L. T.
Ordinate systems for quadratures similar to Tchebycheff's ordinates (D) ... A-109

FLINT, THOMAS
Design and performance of high-pressure axial-flow fan ... 383

FLOW METERS. *See also* CURRENT METERS
Nozzles for (D) ... 251

FLOW OF AIR. *See also* ORIFICES, Intake
Fan discharge ducts (D) ... 346
Measurement, fan inlet and discharge ducts ... 429
Pulsating (D) ... 188

FLOW OF FLUIDS
Research on, review of ... A-28
Turbulent, caused by jets ... A-87

FLOW OF LIQUIDS
Viscous, formula for ... 181

FLOW OF STEAM
Nozzles or blades ... 153, (D) 555

FLOW OF WATER
Current meters for ... 59
Measurement, current meters for ... 227
Measurement, Gibson method of ... 203, (D) 213
Measurement, photoflow method of ... 273

FLUE DUST
Dispersion from chimneys ... 5

FLUID MECHANICS. *See* RESEARCH, Fluid mechanics

FLUID METERS. *See* FLOW METERS

FLUIDS
Properties ... 193
Viscous, laminar flow in ... A-28

FOLSOM, RICHARD G.
Calibration of rounded-approach orifices (D) ... 247
Design and performance of an axial-flow fan (D) ... 344
Propeller pumps ... 197

FOX, ALFRED W.
Cooperation between industrial and public-utility companies in generating steam and electricity (D) ... 20

FRAY, WILLIAM
Obituary of ... RI-49

FREEMAN SCHOLARSHIP COMMITTEE, A.S.M.E.
Members of ... RI-7

FREEMAN SCHOLARSHIPS. *See* JOHN R. FREEMAN TRAVEL SCHOLARSHIPS

FREIGHT TRAINS. *See* RAILROAD TRAINS

FRENCH, DAVID WILLIS
Obituary of ... RI-49

FRICK, A. O.
Obituary of ... RI-50

FRICTION
Bearings ... 169
Fluid, bearing design and ... A-59

FRITZ MEDAL BOARD OF AWARD
A.S.M.E. representatives on ... RI-29

FROCHT, MAX M.
Factors of stress concentration photoelastically determined ... A-67

FRY, ALFRED BROOKS
Obituary of ... RI-50

FRY, LAWFORD H.
Tractive effort of steam locomotives (locomotive ratios—II) (D) ... 366

FUELS DIVISION, A.S.M.E.
Committees of ... RI-9

FULLER, GEORGE WARREN
Obituary of ... RI-51

FURNACES, BOILER. *See* BOILER FURNACES

G

GAFFERT, G. A.
Discussion on paper by ... 141

GAGES
Interferometer strain, use in photoelastic analysis ... A-99

GAHNKIN, V. G.
Ten years of stoker development at Hudson Avenue (D) ... 377

GANTT GOLD MEDAL
Report on ... RI-115

GANTT MEDAL BOARD OF AWARD
A.S.M.E. representatives on ... RI-29

Gases
Viscosity 193
Gates, Thayer Prescott
Obituary of RI-51
Gause, Frederick Taylor
Obituary of RI-52
Gear Teeth
Failure by pitting A-49
Geier, Frederick August
Obituary of RI-52
George Westinghouse Bust Committee, A.S.M.E.
Members of RI-8
Report of RI-110
George Westinghouse Memorial Committee, A.S.M.E.
Members of RI-8
Gershberg, Joseph
Test performance of Hudson Avenue's most recent steam-generating units (D) 267
Geschelin, Joseph
Discussion on paper by 258
Gibson, N. R.
Experimental and practical experience with the Gibson method of water measurement 213
Giesl-Gieslingen, A.
Tractive effort of steam locomotives (locomotive ratios—II) (D) 362
Goff, John A.
Review of recent research in thermodynamics A-65
Technische Thermodynamik (BR) A-38
Thermodynamics (BR) A-160
Goodier, J. N.
Rayleigh's principle (BR) A-76
Goodwin, Harold Lothrop
Obituary of RI-52
Gordon, C. W.
Experimental investigation of heat adsorption in boiler furnaces 541
Gould, G. B.
Relative grindability of coal (D) 191
Grebe, J. J.
High-pressure steam and binary cycles as means of improving power-station efficiency (D) 143
Green, Samuel Martin
Obituary of RI-52
Greenwood, Joseph Rudd
Obituary of RI-53
Grissinger, Elwood
Obituary of RI-53
Grooves
Stress concentration factors for A-67
Guggenheim Medal Fund
A.S.M.E. representatives on RI-29

H

Hagen, H. F.
Air flow in fan-discharge ducts (D) 346
Discussion on paper by 188
Pulsating air flow (D) 189
Hahn, Conrad Velder
Obituary of RI-53
Hailes, Theodore Commodore, Jr.
Obituary of RI-54
Hall, Leo H.
Elastic properties of steel at high temperatures (D) 227
Halloran, Francis Edward
Obituary of RI-54
Hankin, John
Obituary of RI-54
Hardgrove, R. M.
Discussion on paper by 191
Relative grindability of coal (D) 190
Hardie, P. H.
Discussion on paper by 265
Pulsating air flow (D) 188
Harkins, H. Drake
Cooperation between industrial and public-utility companies in generating steam and electricity 11
Harmonic Analysis
Fischer Hinnen method A-137
Ordinate-type method A-137
Hartman, F. V.
Collapse by instability of thin cylindrical shells under external pressure (D) 261
Hartness, James
Obituary of RI-54
Hawkins, G. A.
Discussion on paper by 256
Viscosity of water and superheated steam 395
Heat Absorption. *See* Boiler Furnaces, Heat absorption in
Heat Conduction
Solid circular cylinders, partial-differential equation for A-96
Heat Transfer
Boiler furnaces 105
Heat Transmission
Radiation from nonluminous flames 463
Heck, R. C. H.
Flow distribution in forced-circulation once-through steam generators (D) 257
Helander, Linn
Discussion on paper by 555
Steam-turbine leaving losses and vacuum correction 153
Helical Springs. *See* Springs, Helical
Hendrickson, G. A.
High-pressure steam and binary cycles as means of improving power-station efficiency (D) 142
Hennessy, William Joseph
Obituary of RI-56
Henry Robinson Towne Lecture
New pioneers on a new frontier RI-77
Henry, William Thomas
Obituary of RI-56
Hersey, Mayo D.
Effects of side leakage in 120-degree centrally supported journal bearings (D) 135
Hespe, Frederick William
Obituary of RI-56
Hess, Frank Joseph
Obituary of RI-56
Hewlett, Edward Marriott
Obituary of RI-56
Hirshfeld, C. F.
Cooperation between industrial and public-utility companies in generating steam and electricity (D) 20
Hirt, Louis Joseph
Obituary of RI-57
Hobbs, J. C.
Cooperation between industrial and public-utility companies in generating steam and electricity (D) 21
Hodkinson, B.
Leakage of steam through labyrinth seals (D) 446
Holl, D. L.
Theory of elasticity (BR) A-38
Holley Medal
Recipients of RI-31
Statement about RI-30
Holmes Safety Association
A.S.M.E. representative on RI-29
Holt, Pliny Eastman
Obituary of RI-57
Holtwood Hydraulic Laboratory
Cavitation tests at 455
Honorary Members, A.S.M.E.
List of RI-32
Honorary Membership, A.S.M.E.
Statement about RI-30
Honors Committee, A.S.M.E. ....
Members of RI-8
Hoover Medal
Report on RI-115
Hoover Medal Board of Award
A.S.M.E. representatives on RI-29
Hopping, E. L.
Pulverized fuel-burning experiences at Buzzard Point Station (D) 376
Horger, O. J.
Effect of surface rolling on fatigue strength of steel A-128
Fatigue of shafts at fitted members, with related photoelastic analysis (D) A-71
Hottel, H. C.
Radiation from nonluminous flames 463
Hovgaard, William
Collapse by instability of thin cylindrical shells under external pressure (D) 260
Stresses in three-dimensional pipe bends. 401
Howarth, H. A. S.
Discussion on paper by 355
Loading and friction of thrust and journal bearings with perfect lubrication 169
Hudson Avenue Station
Stoker development at .... 49, (D) 376
Hudson-Orrok Formula
Boiler furnace radiation 105
Huestis, Bronson Lovett
Obituary of RI-58
Humidity
Calculations of .... 229, (D) 557
Hunsaker, J. C.
Aerodynamic theory .. (BR) A-78, (BR) A-156
Conjugate functions for engineers (BR) .. A-119
Progress report on cavitation research at Massachusetts Institute of Technology 423
Theory of functions (BR) A-117
Hutchinson, E. C.
Cooperation between industrial and public-utility companies in generating steam and electricity (D) 22
Hydraulic Division, A.S.M.E.
Committees of RI-9
Hydraulic Turbines
Cavitation testing 455
Hydrodynamics
Theory and design applications, research on A-153
Torsion analogy A-46
Hydroelectric Power Plants
Water measurements in 303

I

Index to Committees, A.S.M.E. RI-33
Industries, Education and Training for, Committee, A.S.M.E.
Members of RI-6
Instrument Control. *See* Regulators
Instruments
Electromagnetic micrometer A-62
Internal-Combustion Engines
Lubrication, research on A-153
International Congresses of Photography, 1935
A.S.M.E. representative to RI-29
International Electrotechnical Commission
A.S.M.E. representatives on RI-29
Report of A.S.M.E. representatives on .. RI-114
Ippen, A. J.
Review of recent research work in fluid mechanics A-28
Iron and Steel Division, A.S.M.E.
Executive committee of RI-9
Irwin, K. M.
Cooperation between industrial and public-utility companies in generating steam and electricity (D) 22

J

Jacobsen, L. S.
Torsional stresses in shafts having grooves or fillets A-154
Jasper, T. McL.
Collapse by instability of thin cylindrical shells under external pressure (D) 262
Resistance of materials (BR) A-158
Jennings, Charles H.
Metallic arc welding (BR) A-158
Welded structures (BR) A-75
Jets. *See* Flow of Fluids, Turbulent
John Fritz Medal
Report on RI-115
John Fritz Medal Board of Award
A.S.M.E. representatives on RI-29
John R. Freeman Travel Scholarship
Recipients of RI-32
Statement about RI-30
Johnson, W. E.
Method of balancing reciprocating machines A-81
Johnson, W. W.
Discussion on paper by 251
Fluid-meter nozzles (D) 254
Jones, Edward Walter
Obituary of RI-58
Joseph A. Holmes Safety Association
A.S.M.E. representative on RI-29
Journal Bearings. *See* Bearings, Journal
Junior Award
Recipients of RI-31
Statement about RI-30
Junior Participation Committee, A.S.M.E.
Members of RI-8
Report of RI-109

Jupp, Alfred John
Obituary of ... RI-58

K

Kaighn, Herbert Eugene
Obituary of ... RI-58
Kameros, S.
Application of elastic-point theory to piping stress calculations ... 165
Karel, Frederick
Obituary of ... RI-58
Karelitz, George B.
Current practice in pressures, speeds, clearances, and lubrication of oil-film bearings (D) ... 357
Discussion on paper by ... A-109
Higher mathematics for engineers and physicists (BR) ... A-159
Ordinate systems for quadratures similar to Tchebycheff's ordinates ... A-12
Recent research work in lubrication ... A-153
Kearney, Edward J.
Obituary of ... RI-59
Keller, C.
Design and performance of an axial-flow fan (D) ... 343
Kent, Robert T.
Current practice in surface broaching (D) 259
Keough, William Thomas
Obituary of ... RI-59
Keppler, P. W.
Ten years of stoker development at Hudson Avenue (D) ... 378
Test performance of Hudson Avenue's most recent steam-generating units (D) 265
Kerr, H. J.
Flow distribution in forced-circulation once-through steam generators (D) ... 256
Kerr, S. Logan
Research investigation of current-meter behavior in flowing water ... 295
Kimball, A. L.
Anecdotal history of the science of sound (BR) ... A-156
Noise (BR) ... A-158
Kingsbury, Albert
Effects of side leakage in 120-degree centrally supported journal bearings (D). 137
Kleffel, H. E.
Economics of preheated air for stokers (D) ... 354
Kleinhans, Schuyler
Seaplane design (BR) ... A-77
Kuethe, Arnold M.
Investigations of turbulent mixing regions formed by jets ... A-87
Kuhns, Austin
Current practice in pressures, speeds, clearances, and lubrication of oil-film bearings (D) ... 355

L

Labyrinths
Leakage through, in steam turbines ... 115, (D) 445
Langer, B. F.
Lateral oscillations of rail vehicles ... 481
Larson, E. L.
Design of light-weight trains (D) ... 128
Lawrence, H. F.
Test performance of Hudson Avenue's most recent steam-generating units (D) 269
Lawrence, James Walter
Obituary of ... RI-59
Lawrence, Russell Ellsworth
Obituary of ... RI-59
Lee, William States
Obituary of ... RI-59
Leffingwell, William Henry
Obituary of ... RI-60
Leonard, Sylvester Maurice
Obituary of ... RI-60
Lessells, John M.
Science and social needs (BR) ... A-159
Lewis, Frank M.
Method of harmonic analysis ... A-137
Library Committee, A.S.M.E.
Members of ... RI-6
Report of ... RI-98
Life Membership, A.S.M.E.
Statement about ... RI-30
Lift. *See* Airfoils, Maximum-lift coefficient of
Lipetz, A. I.
Discussion on paper by ... 362
Liquids
Viscosity ... 193
Local Sections, A.S.M.E.
Executive committees of ... RI-12
Nominating committee groups of ... RI-8
Local Sections Committee, Standing, A.S.M.E.
Members of ... RI-5
Report of ... RI-91
Locomotives. *See also* Railroad Trains,
Lateral oscillations of
High-speed, design requirements of ... 485
High-speed, safe operation of ... 471
Oscillation of ... 471
Ratios (D) ... 362
Steam rate and indicated horsepower of. 495
Steam, tractive effort of. *See* Locomotives, Steam, Performance
Locomotives, Steam
Performance ... 335, (D) 362
Lorsch, Edwin S.
Obituary of ... RI-60
Lubrication
Bearings ... 169, (D) 355
Hydrodynamical theory of, application to journal bearing design ... A-121
Journal bearings (D) ... 135
Pitting dependent on ... A-51
Research on ... A-153
Sleeve bearings ... A-59
Lyman, James
Obituary of ... RI-61
Lynch, Walter Arnold
Obituary of ... RI-61

M

Machine Elements
Fatigue studies of ... A-1
Machine Shop Practice Division, A.S.M.E.
Committees of ... RI-9
Madison, R. D.
Air flow in fan-discharge ducts (D) ... 349
Influence of bends or obstructions at fan-discharge outlet on performance of centrifugal fans (D) ... 244
Intake orifice and proposed method for testing exhaust fans (D) ... 374
Main Award. *See* Charles T. Main Award
Mallory, B. C.
Economics of preheated air for stokers (D) ... 355
Management Division, A.S.M.E.
Committees of ... RI-10
Mann, Nelson Theodore
Obituary of ... RI-61
Manual of Practice Committee, A.S.M.E.
Members of ... RI-8
Report of ... RI-110
Manual on Citizenship Committee, A.S.M.E.
Members of ... RI-7
Marks, Lionel S.
Design and performance of high-pressure axial-flow fan ... 383
Discussion on paper by ... 189, 244, 343, 346
Measurement of air flow in fan inlet and discharge ducts ... 429
Pulsating air flow (D) ... 189
Marmont, E. Leonard
Pitting due to rolling contact (D) ... A-111
Marsh, T. A.
Calculation of dispersion of flue dust and cinders from chimneys (D) ... 9
Marshall, Norman Isaac
Obituary of ... RI-61
Marshall, William Crosby
Obituary of ... RI-62
Massachusetts Institute of Technology
Cavitation research at ... 423
Photoelastic analysis methods at ... A-99
Materials Handling Division, A.S.M.E.
Executive committee of ... RI-10
Mattice, Royal
Obituary of ... RI-62
Maugh, L. C.
Steel structures research (BR) ... A-75
Maulbetsch, John L.
Fatigue of shafts at fitted members, with related photoelastic analysis (D) ... A-71
Thermal stresses in plates ... A-141
Maury, Dabney Herndon
Obituary of ... RI-62
Max Toltz Loan Fund
Statement about ... RI-30
McCarty, Richard Justin
Obituary of ... RI-63
McGeorge, John
Obituary of ... RI-63
McGivern, J. G.
On hydrodynamic analogy of torsion ... A-46
McMillan, Daniel George
Obituary of ... RI-63
Mechanical Engineering
Index to ... RI-125
Mechanical Standards Advisory Council
A.S.M.E. representative on ... RI-29
Meetings
A.S.M.E., 1935, papers presented at ... RI-121
Meetings and Program Committee. A.S.M.E.
Members of ... RI-5
Report of ... RI-89
Mellor, Hiram Lees
Obituary of ... RI-63
Melville Medal
Recipients of ... RI-31
Statement about ... RI-30
Member Relations Committee, A.S.M.E.
Members of ... RI-7
Membership Committee, A.S.M.E.
Members of ... RI-5
Report of ... RI-96
Memorial Notices
A.S.M.E. members ... RI-37
Merrick, Edgar Hamilton
Obituary of ... RI-63
Messer, Vladimir Victor
Obituary of ... RI-63
Metals, Bearing. *See* Bearing Metals
Metals Cutting
Surface broaching (D) ... 258
Metals, Fatigue
Effect of surface rolling on ... A-128
Meters, Current. *See* Current Meters
Micrometers
Electromagnetic for determining shaft positions ... A-62
Mid-Continent Office, A.S.M.E.
Location of ... RI-12
Midwest Office, A.S.M.E.
Location of ... RI-12
Mikina, S. J.
Ordinate systems for quadratures similar to Tchebycheff's ordinates (D) ... A-109
Miller, E. F.
Collapse by instability of thin cylindrical shells under external pressure (D) ... 260
Miller, Harold R.
Obituary of ... RI-64
Millikan, Clark B.
Discussion on paper by ... 192
Theoretical investigation of maximum-lift coefficient ... A-21
Milne, James
Obituary of ... RI-64
Mitchell, Guy Köchling
Obituary of ... RI-64
Mitchell, Robert Lewis, Jr.
Obituary of ... RI-64
Mitereff, Sergei D.
Principles underlying rational solution of automatic-control problems ... 159
Moments of Inertia. *See* Stresses, Pipe lines
Moody, Lewis F.
Progress in cavitation research at Princeton University ... 425
Moreland, Lester A.
Obituary of ... RI-64
Moreton, George William
Obituary of ... RI-65
Morrin, Thomas
Obituary of ... RI-65
Moss, Sanford A.
Calibration of rounded-approach orifices (D) ... 246
Discussion on paper by ... 251
Fluid-meter nozzles (D) ... 254
Pulsating air flow (D) ... 188

Moultrop, I. E.
Economics of preheated air for stokers (D) ... 355
Ten years of stoker development at Hudson Avenue (D) ... 378

Mousson, J. M.
Water gaging for low-head units of high capacity ... 303

Muir, James F.
Cooperation between industrial and public-utility companies in generating steam and electricity (D) ... 23

Mulligan, James E.
Division of load among generating units for minimum cost ... 123

Mullikin, H. F.
Evaluation of effective radiant heating surface and application of Stefan-Boltzman Law to heat absorption in boiler furnaces ... 517
Experimental investigation of heat absorption in boiler furnaces ... 541
Review of methods of computing heat absorption in boiler furnaces ... 531

Muncy, Victor Emanuel
Obituary of ... RI-65

Murray, Henry Hotchkiss
Obituary of ... RI-65

N

Nadai, A.
Eine Losung für die Berechnung der biegsamen rechteckigen Platten. (A General Solution for Rectangular Plates.) (BR) ... A-40
Plasticity and solid state of matter (BR) ... A-115
Review of recent papers on plasticity ... A-33

Nagler, Floyd A.
Use of current meters for precise measurement of flow ... 59

Nathan, Alfred
Obituary of ... RI-65

National Bureau of Engineering Registration
A.S.M.E. representative on ... RI-29

National Defense Division, A.S.M.E.
Executive committee of ... RI-10

National Fire Waste Council
A.S.M.E. representatives on ... RI-29

National Management Council
A.S.M.E. representatives on ... RI-30
Report of A.S.M.E. representatives on ... RI-115

National Research Council
A.S.M.E. representatives on ... RI-30
Report of A.S.M.E. representatives on ... RI-115

Necrology
A.S.M.E. members ... RI-37

Needs, S. J.
Discussion on paper by ... 135

Neefus, Harold Van Houten
Obituary of ... RI 66

Nelson, Victor Robert
Obituary of ... RI-66

Neuber, H. P.
Discussion on paper by ... 138

Neutral-Point Theory
Application to pipe calculations ... 165

Newman, Albert B.
Temperature history during constant-rate heating of solid circular cylinder ... A-96

New York State Joint Legislation Committee
A.S.M.E. representatives on ... RI-30

Nobel Prize
A.S.M.E. representative on ... RI-30

Nolde, Fred
Obituary of ... RI-66

Nominating Committee, 1935, A.S.M.E.
Members of ... RI-8

Nordin, Arvid
Obituary of ... RI-66

Norris, A. M.
Discussion on paper by ... 557
Psychrograph ... 229

Noyes, Mason S.
Collapse by instability of thin cylindrical shells under external pressure (D) ... 263
Exact construction of $(\sigma_1 + \sigma_2)$—network from photoelastic observations (D) ... 139

Nozzles
Fluid-meter (D) ... 251

O

Obituaries
A.S.M.E. members ... RI-37

O'Brien, Edward William
Obituary of ... RI-66

O'Brien, M. P.
Design and performance of an axial-flow fan (D) ... 344
Propeller pumps ... 197
Siphon spillways (BR) ... A-159

Officers, A.S.M.E.
For 1934–1935 ... RI-5

Oil and Gas Power Division, A.S.M.E.
Committees of ... RI-10

Oil Engine Plants
Comparative operating costs, 1929–1933 (table) ... 77
Production cost (table) ... 74

Oil Engine Power Cost
1933 report on ... 69

Oil Engines
Engine details and operating information (table) ... 82

Oiliness
Research on ... A-153

Olive, T. R.
Classification of drying, including graphical analysis of air drying as developed abroad ... 41

Ordinate Systems
Calculation by standard parabolas ... A-14
Rules for calculating ... A-12, (D) A-109
Trapezoidal ... A-12

Orifices
Air measurement with fans ... 431
Intake (D) ... 374
Rounded-approach, calibration of (D) ... 245

Ormondroyd, J.
Atomic structure (BR) ... A-76
From Galileo to cosmic rays (BR) ... A-79
Geometry (BR) ... A-158
New pathways in science (BR) ... A-116
Recent research work in vibration and sound ... A-103
Story of energy (BR) ... A-120
Through space and time (BR) ... A-80

Orrok, Geo. A.
Cooperation between industrial and public-utility companies in generating steam and electricity (D) ... 25

Oscillations. *See* Locomotives, Oscillation of; Railroad Trains, Lateral oscillations of

Otis, Spencer
Obituary of ... RI-67

Ott, L. A.
Observations on use of current meters for precise flow measurement ... 227

P

Page, George Bailey
Obituary of ... RI-67

Papers
Presented at A.S.M.E. meetings, 1935 ... RI-121

Past-Presidents, A.S.M.E.
List of ... RI-32

Pearce, E. S.
Current practice in pressures, speeds, clearances, and lubrication of oil-film bearings (D) ... 356

Peck, C. E.
Airfoil-type propeller blower for ventilating electrical machines ... 417
Influence of bends or obstructions at fan-discharge outlet on performance of centrifugal fans (D) ... 245

Pei, C. P.
Locomotive tractive effort in relation to speed and steam supply ... 335

Perkins, George Edward
Obituary of ... RI-67

Perkins, George Simeon
Obituary of ... RI-67

Perkins, John McClary
Obituary of ... RI-67

Peterson, A.
Air flow in fan-discharge ducts (D) ... 349

Peterson, R. E.
Discussion on paper by ... A-69
Fatigue of shafts at fitted members, with a related photoelastic analysis ... A-1

Petroleum Division, A.S.M.E.
Committees of ... RI-10

Photoelasticity
Fatigue studies of machine elements ... A-1, (D) A-69
Gravity stress measurement by ... A-41
Interferometer strain gage application in ... A-99
Stress analysis by (D) ... 138
Stress concentration factors determined by ... A-67
Thermal-stress measurement by ... A-45

Photography, International Congresses of, 1935
A.S.M.E. representative to ... RI-29

Pigott, R. J. S.
Calibration of rounded-approach orifices (D) ... 247
Current practice in pressures, speeds, clearances, and lubrication of oil-film bearings (D) ... 359
Fluid-meter nozzles (D) ... 253

Pipe Bends
Three-dimensional, stresses in ... 401

Pipe Lines
Stresses ... 165

Pitot Tubes
History of design and use ... 281
Proposed standard ... 430
Water measurement research with ... 295
Water measurement with ... 281
Water measurement with, photoflow method of ... 273
Water velocity measurements with ... A-87

Pitting. *See* Rolling, Pits caused by contact

Plasticity
Mechanism of slip in ... A-33
Volcanic action and ... A-34

Plates
Curved, buckling of ... A-17
Flat, boundary layer along ... A-22
Thermal stresses in ... A-141

Poisson's Ratio
Stress determination by ... A-99

Policies and Budget Committee, A.S.M.E.
Members of ... RI-8
Report of ... RI-110

Poole, Arthur French
Obituary of ... RI-68

Potter, A. A.
Discussion of paper by ... 256
Viscosity of water and superheated steam ... 395

Potter, William Bancroft
Obituary of ... RI-68

Powell, J. A.
Cooperation between industrial and public-utility companies in generating steam and electricity (D) ... 25

Power
Interchange of ... 11
Oil-engine, 1933 report on cost of ... 69

Power Division, A.S.M.E.
Executive committee of ... RI-10

Power Generation
Cycles (D) ... 141
Fundamental economics of (D) ... 28
Industrial and public-utility cooperation in generation of ... 11
Load division among several units ... 123

Power Plant Engineering
Turbine-blade deposits ... 447
Viscosity of water and superheated steam ... 395

Power Plants
Hudson Avenue, Brooklyn, stoker development at ... 49

Power Test Codes
Published by A.S.M.E. in 1935 ... RI-121

Power Test Codes Committee, Standing, A.S.M.E.
Members of ... RI-6
Report of ... RI-104

Power Test Codes Committees, Technical, A.S.M.E.
Members of ... RI-26

Power Transmission *See* Chain Drives

Pownall, W. A.
Tractive effort of steam locomotives (locomotive ratios—II) (D) ... 367

Practice Manual Committee, A.S.M.E.
Members of ... RI-8

Pratt, H. R.
Discussion on paper by (D) ... 244

President, A.S.M.E.
Annual address of ... RI-77

Press Fits. *See* Fatigue, Shafts at fitted members

Priester, G. C.
Collapse by instability of thin cylindrical shells under external pressure (D)..... 259
Princeton University
Cavitation research at................. 425
Printing Industries Division, A.S.M.E.
Committees of......................RI-11
Professional Conduct Committee, A.S.M.E.
Members of.......................... RI-6
Report of.........................RI-106
Professional Divisions Committees, A.S.M.E.
Members of......................... RI-9
Professional Divisions Committee, Standing, A.S.M.E.
Members of.......................... RI-5
Report of..........................RI-94
Professional Engineers Committee on Unemployment
A.S.M.E. representatives on...........RI-30
Prosser, Thomas
Obituary of.........................RI-69
Proutt, Frederick George
Obituary of.........................RI-69
Psychrographs. See Air Conditioning, Psychrographs
Public Affairs Committee, A.S.M.E.
Members of......................... RI-8
Report of.........................RI-110
Publications, A.S.M.E.
Indexes to.........................RI-121
Publications Committee, Special, A.S.M.E.
Members of......................... RI-8
Publications Committee, Standing, A.S.M.E.
Members of.......................... RI-5
Report of..........................RI-89
Publications, Technical Committees
List of............................ 123
Pulverized Coal
Burning at Buzzard Point Station (D)............................ 374
Pumps
Propeller, design of................... 197
Purcell, T. E.
Economics of preheated air for stokers (D).............................. 354
Ten years of stoker development at Hudson Avenue (D).................... 378
Pure Air Committee, A.S.M.E.
Members of......................... RI-9
Representatives of.................. RI-9
Putnam, J. A.
Calibration of rounded-approach orifices (D)............................ 247

Q

Quadratures
Calculation of by different systems ....A-12, (D) A-109

R

Radiant Heating Surface
Boiler furnaces....................... 517
Radiation. See also Boiler Furnaces, Heat absorption in; Boiler Furnaces, Radiation
Nonluminous flames................... 463
Railroad Division, A.S.M.E.
Committees of.......................RI-11
Railroad Locomotives. See Locomotives
Railroad Operation
High-speed locomotives................ 471
Railroad Rolling Stock
Draft gear action in long trains......... 317
Railroad Tracks
Effect of lateral oscillation of train on.................................. 481
Railroad Trains. See also Locomotives, Oscillation of
Lateral oscillations of............... 481
Light weight, design of (D)............. 128
Long, draft gear action in.............. 317
Rails
Effect of lateral oscillation of train on.................................. 481
Railway Axles. See Axles, Railroad cars
Randall, Guy B.
Cooperation between industrial and public-utility companies in generating steam and electricity (D)........... 26
Rathbone, Thomas C.
Dynamic balancing of rotating machinery in the field (D).................... 140
Mathematical solution of rotor-balancing problem (D)....................... 240
Raub, J. H.
Discussion on paper by (D)........... 244
Reciprocating Machines
Method of balancing................. A-81
Registration of Engineers Committee, A.S.M.E.
Members of......................... RI-8
Regulators
Automatic.......................... 159
Relations With Colleges Committee, A.S.M.E.
Members of......................... RI-5
Report of..........................RI-96
Replogle, Mark Anthony
Obituary of.........................RI-69
Representatives on Other Activities
A.S.M.E............................RI-29
Boiler Code......................... RI-7
Power Test Codes...................RI-27
Professional Divisions............... RI-9
Research...........................RI-21
Safety.............................RI-28
Standardization....................RI-25
Research
Cavitation. See Cavitation, Research
Fluid mechanics, review of.............A-28
Lubrication.........................A-153
Vibrations..........................A-103
Research Committees, Special, A S.M.E.
Members of.........................RI-20
Research Committee, Standing, A.S.M.E.
Members of......................... RI-6
Report of..........................RI-98
Research Publications
A.S.M.E., 1935.....................RI-121
Reynolds, Myron B.
Obituary of.........................RI-60
Rheingans, W. J.
Photoflow method of water measurement.............................. 273
Rhys, C. O.
Collapse by instability of thin cylindrical shells under external pressure (D)..... 260
Rice, Calvin Winsor
A.S.M.E. Council tribute to...........RI-39
Contribution to international friendliness..............................RI-40
Life of............................RI-37
Portrait of.........................RI-36
Rice Memorial Committee, A.S.M.E.
Members of......................... RI-8
Report of.........................RI-110
Rice Memorial Lecture
First...............................RI-40
Ricketts, Palmer Chamberlaine
Obituary of.........................RI-70
Roark, R. J.
Strength and stiffness of cylindrical shells under concentrated loading.....A-147
Robinson, Ernest L.
Steam-turbine leaving losses and vacuum corrections (D).................... 555
Robinson, William
Obituary of.........................RI-71
Roller, Chain Drives
Rating of.....................97, (D) 443
Rolling
Pits caused by contacts.......A-49, (D) A-110
Stresses due to metal-to-metal contact ....A-54, (D) A-110
Surface, effect on fatigue strength of steel.............................. A-128
Romaine, Millard
Current practice in surface broaching (D).............................. 258
Roop, W. P.
Collapse by instability of thin cylindrical shells under external pressure (D)..... 260
Ross, M. D.
Airfoil-type propeller blower for ventilating electrical machines.............. 417
Rotors
Balancing...................(D) 140, (D) 239
Royle, Vernon
Obituary of.........................RI-71
Rubenkoenig, H.
Tractive effort of steam locomotives (locomotive ratios—II) (D).......... 366
Ryan, James J.
Dynamic balancing of rotating machinery in the field (D).................... 140
Mathematical solution of rotor-balancing problem (D)........................ 241

S

Saegmuller, George Nicholas
Obituary of.........................RI-72
Safety Committee, Standing, A.S.M.E.
Members of......................... RI-6
Report of.........................RI-106
Safety Committees, Technical, A.S.M.E.
Members of.........................RI-27
St. Louis Medal. See Spirit of St. Louis Medal
Sanderson, Edwin Joel
Obituary of.........................RI-72
Schmarje, C. F.
Radiation intensities and heat transfer by radiation in boiler furnaces........... 105
Scorah, R. L.
Leakage of steam through labyrinth seals (D).............................. 445
Seals
Labyrinth, leakage through, in steam turbines....................115, (D) 445
Seidle, Norman Robert
Obituary of.........................RI-72
Selaya, M. J.
Economics of preheated air for stokers (D).............................. 353
Shafts
Fatigue tests (D).................... A-69
Fatigue tests, Westinghouse Research Laboratories....................... A-1
Strength at fitted members............ A-1
Stress lines due to torsion in.......... A-47
Torsional stresses in................A-154
Shamberger, J. P.
Lateral oscillations of rail vehicles....... 481
Sheldon, Lucian A.
High-pressure steam and binary cycles as means of improving power-station efficiency (D)...................... 141
Shells
Cylindrical, concentrated loading of.....A-147
Ships
Displacement of..................... A-12
Resistance.......................... A-29
Stability of......................... A-12
Simpson's Ordinates
Rule for calculating.................. A-12
Skinkle, W. B.
Cooperation between industrial and public-utility companies in generating steam and electricity (D)........... 27
Slag
Calculation of effect on boiler tubes..... 527
Sleeve Bearings. See Bearings, Sleeve
Sloman, H. J.
Discussion on paper by............... 190
Smith, Ed S., Jr.
Air flow in fan-discharge ducts (D)...... 348
Calibration of rounded-approach orifices (D)............................ 246
Discussion on paper by............... 249
Smith, George Hotchkiss
Obituary of.........................RI-72
Smith, J. F. Downie
Air flow in fan-discharge ducts (D)...... 347
Discussion on paper by............... 245
Smith, Ronald B.
Calibration of rounded-approach orifices (D)............................ 246
Fluid-meter nozzles (D)............... 251
Smith, V. C.
Radiation from nonluminous flames..... 463
Smithwick, Nathaniel Austin
Obituary of.........................RI-72
Smoke Abatement
Calculation of dispersion of flue dust from chimneys........................... 5
Soderberg, C. Richard
Cooperation between industrial and public-utility companies in generating steam and electricity (D)........... 27
Mechanical vibrations (BR)............ A-39
Torsional vibration (BR).............. A-39
Working stresses....................A-106
Solberg, H. L.
Discussion on paper by............... 256
Viscosity of water and superheated steam.............................. 395

SOMERS, W. E.
Definition of quantity-head characteristic of fans 433
SORENSON, ALFRED E.
Progress in cavitation research at Princeton University 425
SOUNDS
Vibrations and, research on A-103
SPALLING
Theory of (D) A-111
SPERR, W. H.
Discussion on paper by 376
Ten years of stoker development at Hudson Avenue 49
SPIELVOGEL, S. W.
Application of elastic-point theory to piping stress calculations 165
SPIRIT OF SAINT LOUIS MEDAL
Recipients of RI-31
Statement about RI-30
SPIRIT OF ST. LOUIS MEDAL BOARD OF AWARD, A S.M.E.
Members of RI-8
SPRENKLE, R. E.
Calibration of rounded-approach orifices (D) 245
Fluid-meter nozzles (D) 251
SPRINGS
Compression, design data for A-35
Helical, constants for deflections of A-86
Helical, design data for A-35
Helical, improved materials for A-34
Tension, design data for A-35
SPRING WIRE
Bending and torsion fatigue tests of 501
STAHL, E. C. M.
Test performance of Hudson Avenue's most recent steam-generating units (D) 268
STALKER, E. A.
Further experiments on variation of maximum-lift coefficient with turbulence and Reynolds' number (D) 192
STANDARDIZATION COMMITTEE, STANDING, A.S.M.E.
Members of RI-6
Report of RI-102
STANDARDIZATION COMMITTEES, TECHNICAL A.S.M.E.
Members of RI-22
STANDARDS
Published by A.S.M.E. in 1935 .... RI-121
STANDARDS AND CODES, ADVISORY BOARD on, A.S.M.E.
Members of RI-7
Report on RI-107
STANDING COMMITTEES, A.S.M.E.
Members of RI-5
STEFAN-BOLTZMAN LAW
Application to heat absorption in boiler furnaces 517
STEAM
Viscosity of 395
STEAM CYCLES
High pressure and binary (D) 141
STEAM FLOW. *See* FLOW OF STEAM
STEAM GENERATION. *See also* POWER GENERATION
Industrial and public-utility cooperation in 11
STEAM GENERATORS. *See* BOILERS
STEAM LOCOMOTIVES. *See* LOCOMOTIVES, STEAM
STEAM POWER PLANTS
Buzzard Point (D) 374
STEAM TURBINES
Blades, deposits on 447
Losses .... 153, (D) 555
Losses through labyrinth seals .... 115, (D) 445
Vacuum corrections .... 153, (D) 555
STEEL
Cold-working, effect on physical properties of A-128
Effect of surface rolling on fatigue strength of A-128
Elastic properties at high temperatures 1, 272
STEPANOFF, ALEXEY J.
Design and performance of an axial-flow fan (D) 345
STOKERS
Development .... 49, (D) 376
Hudson Avenue Station .... 49, (D) 376
Preheated air for (D) 351
STONE, M.
Film lubrication in sleeve bearings A-59
STONE, THOMAS WADE
Obituary of RI-73
STRAUB, FREDERICK G.
Cause and prevention of turbine-blade deposits 447
STRENGTH OF MATERIALS. *See also* ENDURANCE OF METALS; METALS FATIGUE
Ductile, under different conditions of stress and temperature A-106
Fatigue tests (D) A-69
STRESSES. *See also* SPRINGS, Helical, design data for
Analysis, photoelastic method of (D) 138
Concentration factors photoelastically determined A-67
Cylindrical shells .... (D) 144, (D) 259
Cylindrical shells under concentrated loading A-147
Determination by photoelastic methods A-99
Gravity, measurement of A-41
$(\sigma_1 + \sigma_2)$—network (D) 138
Pipe bends 401
Pipe lines 165
Rolling .... A-54, (D) A-110
Shafts at fitted members .... A-1, (D) A-69
Temperature, two-dimensional, cylindrical bodies A-41
Thermal, in plates A-141
Torsional in shafts having grooves and fillets A-154
Working A-106
STROWGER, E. B.
Experimental and practical experience with the Gibson method of water measurement 213
STRUCKMANN, EDWIN
Obituary of RI-73
STUART, M. C.
Definition of quantity-head characteristic of fans 433
STUDENT AWARDS
Recipients of RI-31
Statement about RI-30
STUDENT BRANCHES, A.S.M.E.
List of RI-18
STURM, R. G.
Collapse by instability of thin cylindrical shells under external pressure (D) 261
STYRI, HAAKON
Pitting due to rolling contact (D) A-112
SUPIOT, CHARLES ANTHONY
Obituary of RI-73

T

TCHEBYCHEFF'S ORDINATES
Comparison with other systems .... A-12, (D) A-109
TECHNICAL COMMITTEE PUBLICATIONS
List of 123
TECHNICAL COMMITTEES, STANDING, A.S.M.E.
Members of RI-6
TECHNICAL COMMITTEES, SPECIAL, A.S.M.E.
Members of... RI-6, RI-20, RI-22, RI-26, RI-27
TECHNOLOGY, ADVISORY BOARD ON, A.S.M.E.
Members of RI-7
Report of RI-106
TEMPERATURE
Stresses due to A-44
TENSILE TESTS
Copper, under high velocities of deformation A-33
TESTING MACHINES
Fatigue of metals A-3
Pitting A-50
TEXTILE DIVISION, A.S.M.E.
Executive committee of RI-11
THEARLE, E. L.
Discussion on paper by 140
Mathematical solution of rotor-balancing problem (D) 239
THERMODYNAMICS
Research in A-65
THERMOSTATS
Buckling of bi-metallic slips of A-19
THIELSCHER, H. G.
Discussion on paper by 374
THOMA, D.
Concerning degree of accuracy of Gibson method of measuring flow of water 203
Discussion on paper by 213
THOMAS, CARROLL D.
Obituary of RI-73
THOMAS, CHARLES W.
Classification of drying, including graphical analysis of air drying as developed abroad 41
THRUST BEARINGS. *See* BEARINGS, THRUST
TICHVINSKY, L. M.
Journal bearing performance A-121
New method of investigating performance of bearing metals (D) 380
TIMOSHENKO, S.
Buckling of flat curved bars and slightly curved plates A-17
TOLTZ FUND. *See* MAX TOLTZ LOAN FUND
TORSION. *See also* SPRING WIRE
Hydrodynamic analogy of A-46
TOWNE, HENRY ROBINSON, LECTURE
New pioneers on a new frontier RI-77
TOWNSEND, J. R.
Hardness of metals and its measurement (BR) A-157
TRAINS. *See* RAILROAD TRAINS
TRANSACTIONS, A.S.M.E.
Depositories for RI-117
Index to RI-134
TRANSMISSION CHAINS
Roller, rating of .... 97, (D) 443
TRILLING, CHARLES
Discussion on paper by 259
TUBES
Rolling 145
TURBINE BLADES. *See* STEAM TURBINES, Blades
Hydraulic. *See* HYDRAULIC TURBINES
Steam. *See* STEAM TURBINES
TURBO-GENERATORS
Cooling 417
TURBULENCE
Prandtl's theory of A-87
Research on A-28
Tollmein's solution for A-87

U

UNDERWOOD, A. F.
Current practice in pressures, speeds, clearances, and lubrication of oil-film bearings (D) 360
UNITED ENGINEERING TRUSTEES, INC.
A.S.M.E. representatives on RI-30
Report of A.S.M.E. representatives on ... RI-115

V

VAN BRUNT, JOHN
Economics of preheated air for stokers (D) 352
VERSE, GUY
Discussion on paper by 272
Elastic properties of steel at high temperatures 1
VIBRATION
Research on A-103
VIESSMAN, WARREN
Psychrograph (D) 557
VISCOMETERS
Lawaczeck use in investigation of viscosity of water and superheated steam 395
VISCOSITY
Absolute and kinematic 193
Lubricant for sleeve bearings A-59
Measurement of, research on A-153
Oil, effect on pitting A-51
Water and superheated steam, power-plant practice 395
VOLCANOES
Secondary cones, location on regular spiral lines A-34
VON KARMAN, TH.
Review of recent research work in fluid mechanics A-28
Theoretical investigation of maximum-lift coefficient A-21
VON MILLER, OSKAR
Obituary of RI-73
VORTEX SYSTEMS
Generation and motion of A-29
VOSE, R. W.
Application of interferometer strain gage in photoelasticity A-99

W

WADDELL, JOSEPH ADDISON, JR.
Obituary of RI-74

WAHL, A. M.
Discussion on paper by A-69
Fatigue of shafts at fitted members, with a related photoelastic analysis A-1
Helical compression and tension springs.. A-35

WALTERS, VINCENT L.
Obituary of RI-75

WARD, N. F.
Reliability of aircraft welds 389

WARNER MEDAL. *See* WORCESTER REED WARNER MEDAL

WASHINGTON AWARD
A.S.M.E. representatives on RI-30
Report of A.S.M.E. representatives on RI-115

WATER
Viscosity 193, 395

WATER-AIR MIXTURES
Chart of 45

WATER FLOW. *See* FLOW OF WATER

WATER MEASUREMENT. *See* CURRENT METERS; FLOW OF WATER, Current meters for; FLOW OF WATER, Measurement; PITOT TUBES; WEIRS, V-notch

WATTENDORF, F.
Hydro- and aeromechanics (BR) A-118

WAY, STEWART
Discussion on paper by A-110
Fatigue of shafts at fitted members, with related photoelastic analysis (D) A-69
New theory for buckling of thin cylinders under axial compression and bending (D) 144
Pitting due to rolling contact A-49

WEIBEL, E. E.
Correlation of spring-wire bending and torsion fatigue tests 501

WEIBEL, E. E. *(continued)*
Exact construction of ($\sigma_1 + \sigma_2$)—network from photoelastic observations (D) 138
Photoelasticity (BR) A-118

WEIRS
V-notch (D) 249

WEISSELBERG, ARNOLD
Classification of drying, including graphical analysis of air drying as developed abroad 41
Psychrograph (D) 558

WELDING
Fusion, aircraft 389

WELDS
Aircraft 389

WESKE, JOHN R.
Discussion on paper by 343

WESSTROM, D. B.
Collapse by instability of thin cylindrical shells under external pressure (D) 261

WESTERN SOCIETY OF ENGINEERS, WASHINGTON AWARD
A.S.M.E. representatives on RI-30
Report of A.S.M.E. representatives on RI-115

WESTINGHOUSE BUST COMMITTEE, A.S.M.E.
Members of RI-8

WESTINGHOUSE MEMORIAL COMMITTEE, A.S.M.E.
Members of RI-8

WHEELER, FRANK ROSE
Obituary of RI-75

WHIPPLE, GEORGE E.
Obituary of RI-75

WHITE, W. M.
Photoflow method of water measurement 273

WHITFIELD, R.
Discussion on paper by 374

WIKANDER, O. R.
Draft-gear action in long trains 317

WILLIAMS, ARTHUR
Measurement of steam rate and indicated horsepower of locomotives 495

WILLISTON, WILLIAM HIGH
Obituary of RI-75

WINDENBURG, DWIGHT F.
Discussion on paper by 259

WIRE
Valve-spring, bending and torsion fatigue tests, correlation of 501

WOHLENBERG METHOD
Boiler furnace radiation 105

WOHLENBERG, W. J.
Experimental investigation of heat absorption in boiler furnaces 541
Review of methods of computing heat absorption in boiler furnaces 531

WOMAN'S AUXILIARY SCHOLARSHIP
Statement about RI-30

WOOD INDUSTRIES DIVISION, A.S.M.E.
Committees of RI-11

WOOD, WALTER
Obituary of RI-75

WORCESTER REED WARNER MEDAL
Recipients of RI-31
Statement about RI-30

WRIGHT, A. M.
Design of light-weight trains (D) 129

Y-Z

YATES, H. G.
Leakage of steam through labyrinth seals (D) 446

YOUNG, E. G.
Locomotive tractive effort in relation to speed and steam supply 335

ZEH, EDMUND WILHELM
Obituary of RI-75

1935 Annual Meeting Papers in This Issue

TRANSACTIONS

of The American Society of Mechanical Engineers

Cause and Prevention of Turbine-Blade Deposits (FSP-57-14) . . . . [illegible]

Cavitation Testing of Model Hydraulic Turbines and Its Bearing on Design and Operation (HYD-57-1) . . . . *L. M. Davis* 455

Radiation From Nonluminous Flames (PRO-57-4) . . . . *H. C. Hottel and V. C. Smith* 463

Safe Operation of [illegible] Locomotives [illegible] . . . . [illegible]

Lateral Oscillations of Rail Vehicles (RR-57-4) . . . . *B. F. Langer and J. P. Shamberger* 481

Measurement of Steam Rate and Indicated Horsepower of Locomotives (RR-57-5) . . . . *Arthur Williams* 495

The Correlation of Spring-Wire Bending and Torsion Fatigue Tests (RP-57-1) . . . . *E. E. Weibel* 501

Evaluation of Effective Radiant Heating Surface and Application of the Stefan-Boltzman Law to Heat Absorption in Boiler Furnaces (RP-57-2) . *H. F. Mullikin* 517

Review of Methods of Computing Heat Absorption in Boiler Furnaces (RP-57-3) . . . . *W. J. Wohlenberg and H. F. Mullikin* 531

An Experimental Investigation of Heat Absorption in Boiler Furnaces (RP-57-4) . . . . *W. J. Wohlenberg, H. F. Mullikin, W. H. Armacost, and C. W. Gordon* 541

Discussion 555
On previously published papers by Linn Helander; and A. M. Norris

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